TP 8399E

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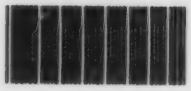

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STUDY OF ACTIVE SUSPENSIONS FOR HEAVY VEHICLES

by

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March 1987
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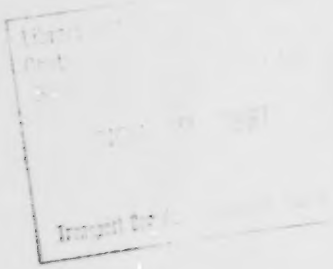

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Un sommaire en français de ce rapport est inclus avant la Table des matières.

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17. Abstract

A literature review and study is described in which the potential usefulness of active suspensions for heavy vehicles is investigated. Current heavy vehicle suspensions and some of their dynamic road-loading characteristics are reviewed. The performance requirements of an active suspension are described as a function of environmental, mechanical and control criteria. The causes of heavy vehicle accidents and the effect of suspension dynamics on these are examined. The infrastructural and operating costs associated with heavy vehicles are reviewed. Road wear mechanisms and the effect of heavy vehicle suspension systems on road damage are presented. The findings of the study indicate that the adoption of an active suspension would likely result in reduced road wear, a lesser involvement of heavy vehicles in accidents, and reduced operating costs.

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17. Résumé

Ce rapport traite d'une étude portant sur le potentiel des suspensions actives pour véhicules à poids lourds. Les configurations mécaniques ainsi que les réactions dynamiques des suspensions existantes sont revues. Le fonctionnement idéal d'une suspension active est décrit selon l'importance de plusieurs critères. La distribution des accidents des poids lourds est revue, surtout en fonction de certains genres de suspensions. Les coûts d'opération du transport par poids lourds et ceux des infrastructures routières sont énumérés. Les sources de l'usure des chaussées sont présentées. Cette étude trouve que l'utilisation d'une suspension active sur les véhicules à poids lourds s'avérerait fort utile puisque celle-ci pourrait réduire l'usure des chaussées, empêcher certains genres d'accidents ainsi que minimiser les coûts d'opérations.

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SUMMARY

Operating economy and operating safety have the greatest influence, other than legal requirements, over the instigation and implementation of design modifications in heavy vehicles. Furthermore, in order for any new feature or component to be successfully introduced into common usage among these vehicles, it must first be shown that it can solve or alleviate a well-recognized problem.

Across Canada, provincial legislation requires the linking of certain axles on heavy vehicle trailers (tandem-type axles). The rationale for such legislation is entirely justified, as the links are an attempt to better distribute wheel loads in order to reduce road wear. However, the presence of such inter-axle links introduces certain compromises in the design of the overall suspension system. In such cases, a suspension designed to equalize static axle loads might actually amplify dynamic road loads under certain conditions.

In order to address the true intent of this type of legislation, which is to reduce the level of road wear as well as to increase operating safety, a review of heavy vehicle suspension characteristics is in order.

HEAVY VEHICLE SUSPENSIONS

Purpose: In general terms, the purpose of a suspension system is to isolate a vehicle's sprung mass from road surface irregularities and to protect its cargo, and/or passengers, from harmful acceleration loadings.

Operating conditions: A typical road vehicle suspension is subject to vertical displacements with amplitudes of zero to ten centimetres (possibly greater on roads in very poor condition) and frequencies typically in the range of zero to ten Hertz (or higher at elevated vehicle speeds).

Heavy vehicles used on paved roads are subject to similar disturbances. However, due to the widely varying loads carried by these vehicles, the dynamic reactions of their suspension systems can exceed those levels which would be desirable in order to minimize road damage.

Classical suspension: The traditional suspension used to isolate road vehicles from bumps and holes has been a spring and damper (present as friction in the spring in some cases) assembly placed between the road and the vehicle. In most cases this is done by suspending each road wheel on a linkage whose vertical displacements and accelerations are moderated by a spring and a damper.

Types of springs and dampers: Many types of springs have been used, such as steel leafs, coils or bars, a constrained gas, a combination of hydraulic fluid and gas under pressure, rubber in shear, and a number of others. Dampers usually take the form of a hydraulic cylinder whose piston has specially designed orifices which restrain the passage of fluid to a predefined rate according to the piston's velocity and/or acceleration.

Several types of heavy vehicle suspensions do not have distinct damping mechanisms. Rather, the friction present in the springs or suspension linkage points serves as a means of absorbing some of the energy introduced into the system.

Suspension displacement: The unsprung mass of the wheel is free, within the spring's design limits, to move up or down as it encounters bumps or depressions in the roadway. As this movement occurs, the damper (or spring friction) absorbs some of the energy present in the system and attempts to prevent the wheel from attaining excessive vertical accelerations.

Stiffness and damping rates: A heavy vehicle's suspension spring rate (or characteristic frequency) is usually fixed, and specified as a function of the maximum expected load to be carried. Under all loading conditions, however, the vehicle must retain a certain level of stability and control. Further, it is desirable to maintain an acceptable ride for the cargo and/or occupants whether the vehicle is full or near-empty. It has proven to be difficult, if not impossible, to arrive at a suspension compromise of the above requirements which would also minimize wheel reactions in such a way as to reduce road wear.

HEAVY VEHICLE SUSPENSION REACTIONS AND ROAD WEAR

Recent research has indicated that a definite link can be established between a suspension's dynamic characteristics and its effects on pavement loading. In order to minimize the level of road damage that is a direct result of this pavement loading, certain studies have proposed suspension design guidelines. As previously noted however, it is quite difficult to satisfy such guidelines while still using traditional heavy vehicle suspension equipment, especially if the differences in mass between the loaded and unloaded conditions are significant.

A modification or control of the typical suspensions' behaviour could reduce its level of pavement loading. Certain air-bag type suspensions lend themselves to the use of dampers which exhibit a variable damping rate as a function of the air pressure in the air-bladders. However, the actual damping range available under such circumstances

is limited, and is not made to vary according to other factors (ie. wheel position, resonance, etc.). In order to establish full control over wheel movements and the ensuing vehicle behaviour, an active suspension is required.

ACTIVE SUSPENSION: ITS CHARACTERISTICS

An active suspension may be described as one which can exert or generate a force independent of the direction and of the velocity of the wheel or suspension unit. Thus, its behaviour can be a function of several variables. In contrast, a passive or semi-active (controlled damping) suspension can only generate a force relative to local motion.

An active suspension, by definition, can change its springing and/or damping rates as a function of varying vehicle load, road conditions or driving styles. Its main purpose is to eliminate undesirable ride frequencies and prevent vehicle roll or pitch. An effective means of sensing and controlling an active suspension's behaviour is through the use of micro-electronics. Sensors can monitor the position, speed and acceleration of each road wheel, accelerometers can measure vehicle conditions, and transducers can gauge the load present at each wheel. According to these inputs, an electronic control unit can determine the appropriate force values (the effective spring and/or damper rates) to be set at the suspension's various wheel locations.

ACTIVE SUSPENSION: ITS ADVANTAGES

The outcomes of the proper installation and control of an active suspension are several. Some of these are advantageous in contrast to a non-active suspension system:

-The spring and damping rates are variable, individually and instantaneously, according to the static and dynamic conditions imposed at each wheel (load, velocity, acceleration, etc.). Among other things, this characteristic can reduce or eliminate axle tramp and its damaging effects on pavement.

-The instantaneously variable spring and damping rates are modified according to the "real-time" dynamics of the vehicle as a whole (such as lateral acceleration, etc.) as well as in relation to outside factors (such as grade severity, lateral banking level, etc.). As such, the suspension conditions which can contribute to incidents such as rollovers, which constitute 7% of all commercial articulated vehicles accidents, can be eliminated.

-Static levelling is automatic, regardless of the position and severity of each dead load, since the required supporting reaction force in the suspension is independent of "spring" deflections or chassis height. Dynamic levelling, a normal characteristic of an active system, enhances stability and controllability by maintaining a level platform and preventing undesirable chassis accelerations.

-The ride height can be operator selectable in case of difficult road conditions (snow laden or rock-strewn, etc.) or non-standard loading dock heights.

-Wheel loadings, cargo weight distribution, vehicle attitude and other types of information can be displayed on a dashboard-located screen. Safety warning functions, such as "unsafe-loading" notices, would warn of possibly hazardous operating conditions and suggest remedial actions to be undertaken, thus contributing to acceptable tire and trailer loadings. A feature might even be developed to evaluate the progressive change in suspension performance characteristics and warn of possible failures in the suspension, tires, brakes or other components.

The most important advantage however, could very well be the effect such a suspension would have on the reduction of road wear. The extent to which this reduction would occur can be inferred by examining the identified causes of pavement damage and how these are related to suspension behaviour.

REDUCTION OF ROAD LOADING BY ACTIVE SUSPENSIONS

Road damage has been suggested to be proportional to the fourth power of the axle weight of any given vehicle. A reduction of less than 16% in the dynamic weight (suspension reaction) would thus result in a 50% reduction in road damage. Studies performed on the experimental use of semi-active suspensions on aircraft have exceeded this percentage reduction of dynamic weight, as compared to a classical suspension.

It is reasonable to assume that a fully active suspension in use on a heavy vehicle would demonstrate a similar reduction, and yield a similar percentage reduction in road damage. This could lead to a lesser requirement in the level of maintenance of roads, or, alternatively, to a greater gross vehicle weight allowance for vehicles equipped with such an active suspension.

There are other aspects of the use of an active suspension which could assist in the effective reduction of pavement loadings. Such a suspension could permit an even sharing of the vehicle's total load among the available axles. Distinct

axles or wheels could follow more faithfully the actual driving surface (road or obstacle) without inducing other wheels or axles to rise away from it. While following this surface, each wheel or axle would continue to support an appropriate share of the overall load, as determined by the control system.

REDUCTION OF ROAD ACCIDENTS WITH ACTIVE SUSPENSIONS

Many elements may be involved in, or contribute to, the occurrence of a heavy vehicle accident. The main contributing factors have been identified as the following: Operator, Vehicle, Environment, Cargo. An active suspension could help to minimize the importance of these factors, thus reducing the overall number of accidents.

Rollovers, for example, account for up to 15 percent of all of the accidents involving heavy vehicles. An active suspension could reduce the incidence of rollovers by enhancing a vehicle's instantaneous roll-resistance.

REDUCTION OF OPERATING COSTS WITH ACTIVE SUSPENSIONS

The costs of providing and maintaining a road infrastructure as well as the costs of operating a heavy vehicle can be reduced by using active suspensions.

The road reaction forces of an active suspension can be quite low in comparison to those of a passive suspension. Controlled experiments have shown that reduced dynamic loads can result in lower road wear which then leads to reduced maintenance costs. Similarly, reduced suspension loads introduced into the vehicle can lead to a longer service life for many components such as the tires, the body and frame, and many others.

Rough road surfaces have been shown to significantly increase the effective rolling resistance imposed to a passing vehicle. A reduction of such losses could enhance fuel economy, and thus minimize operating costs. A properly calibrated active suspension would encounter a lower coefficient of rolling resistance on a rough surface than would a passive suspension, through the selected damping of certain axle and wheel motions.

CONCLUSIONS

The technology for a practical, effective and moderately priced active suspension is now available.

The potential benefits of the use of an active suspension on heavy vehicles are important and widespread. The potential savings in road maintenance as well as in vehicle operating costs could be of major significance to both the private operator and to the community at large.

The design of an optimal, software based controller philosophy is probably more complex than the design of the electronic hardware and the mechanical suspension components of an active suspension. Neither of these tasks, however, are considered to contain major technical obstacles.

RECOMMENDATIONS

The design, implementation and quantitative trial of an active suspension system for heavy vehicles should be performed.

SOMMAIRE

Au Canada, la réglementation provinciale des véhicules routiers exige que les essieux des remorques à poids lourds soient reliés afin d'égaliser leurs charges. Ces règlements sont bien fondés, du moins en ce qui a trait à la répartition des charges statiques, comme dans le cas d'une remorque stationnaire.

Toutefois, de ces interconnexions résultent certains compromis au plan de la conception des suspensions des essieux. En général, ces compromis entraînent des effets négatifs sur les réactions dynamiques des suspensions: la répartition des charges dynamiques d'une utilisation normale sur la route n'est plus égale et, celles-ci peuvent même être amplifiées à cause du gabarit de la suspension.

Afin de mieux rencontrer les objectifs de la réglementation provinciale prônant une usure réduite des chaussées ainsi qu'une sécurité d'opération accrue, il est nécessaire d'entreprendre une revue des caractéristiques des suspensions pour véhicules à poids lourds.

LES SUSPENSIONS DES VEHICULES A POIDS LOURDS.

Raison d'être: La fonction d'une suspension véhiculaire est d'atténuer ou d'éliminer les accélérations et déplacements verticaux imposés à un véhicule et à son contenu. Les sources de ces accélérations et déplacements sont des bosses, des trous, ainsi que d'autres obstacles que l'on retrouve sur les routes.

Echelle d'atténuation requise: La suspension d'une voiture de tourisme doit atténuer des déplacements verticaux ayant des amplitudes de + ou - 10 cm et des fréquences allant de zéro à dix Hertz. Ces déplacements peuvent être plus importants sur des routes accidentées et une fréquence plus élevée peut être rencontrée à une haute vitesse de croisière.

Les véhicules à poids lourds sont également soumis à de semblables déplacements verticaux. Toutefois, en vertu des masses variables portées par ces véhicules, ainsi que de la configuration reliée des suspensions, les réactions dynamiques qui y sont générées dépassent largement le niveau qui permettrait de minimiser la détérioration des chaussées.

Solution classique: Utilisée par tous genres de véhicules routiers, la suspension classique, composée d'un ressort et d'un amortisseur, isole le véhicule de la route. Ordinairement cette combinaison du ressort et de l'amortisseur est placée entre la masse suspendue (véhicule,

tracteur ou remorque) et les masses non-suspendues (roues, essieux et bras de suspension).

Genres de ressorts et d'amortisseurs: Des ressorts de nombreuses configurations sont fabriqués de plusieurs genres de matériaux. On peut retrouver des lames, des barres et des hélices en acier. Au niveau des accumulateurs hydropneumatiques, la chambre compressible permet à l'élasticité du gaz ou de l'air comprimé qui s'y trouve de servir de ressort. D'autre part, on rencontre des masses de caoutchouc soumisent au cisaillement ou à la compression et qui réagissent de façon semi-élastique.

En ce qui a trait aux amortisseurs, ils prennent souvent la forme d'une chambre de compression dont la pression peut s'échapper lentement par l'entremise d'ouvertures prévues à cette fin. On retrouve des vérins équipés de soupapes ou de pistons perforés qui permettent un écoulement à débit prescrit du fluide (air, gaz, liquide hydraulique). Cet écoulement limité empêche une accélération excessive du piston et, de ce fait, réduit la vitesse et l'accélération de déplacement du ressort ainsi que de la suspension. Il prévient également que ce ressort ne soit sujet à certaines fréquences d'oscillations cycliques.

Plusieurs genres de suspensions pour véhicules à poids lourds n'ont pas d'amortisseurs comme tel. Dans ces cas, l'amortissement se fait grâce à la friction entre les composantes de la suspension (par exemple entre les lames des ressorts en acier à plusieurs épaisseurs) ou encore comme perte interne d'énergie au niveau du ressort (un ressort en caoutchouc n'est pas parfaitement élastique).

Déplacement de la suspension: Sujette aux limites de pompage, la masse non-suspendue (roue, pneu, axe, etc.) peut se déplacer verticalement sans occasionner un déplacement équivalent du véhicule. La proportion du déplacement transmise au véhicule par la suspension dépend de son taux d'amortissement ainsi que de la raideur de son ressort.

La raideur et l'amortissement: Le ressort est généralement de raideur fixe ou de raideur progressive à pente fixe, selon le cas. Cette raideur (la fréquence caractéristique du ressort) est spécifiée suite à une étude qui doit déterminée le meilleur compromis entre une tenue de route sécuritaire et une conduite (confort) acceptable pour le transport de biens et de passagers, dont la charge peut varier.

L'amortissement peut être rendu variable en fonction de la vitesse et/ou de l'accélération verticale de la masse suspendue, en utilisant de multiples soupapes à résistance d'ouvertures variées. Toutefois, un amortisseur est conçu selon une échelle restreinte de charges d'utilisation, ce qui le rend inéfficace au delà de celle-ci.

LES REACTIONS DES SUSPENSIONS DES VEHICULES A POIDS LOURDS, ET LA DETERIORATION DES CHAUSSEES.

Depuis quelques années, de nombreuses études ont établi qu'il existe un lien entre le taux de la détérioration de la chaussée et l'intensité des forces dynamiques qui lui sont imposées. D'autres études préconisent la mise en pratique de certains critères de conception au niveau des systèmes de suspensions, afin de réduire l'intensité de leurs réactions dynamiques. Les réactions dynamiques sont d'autant plus importantes dans le cas des véhicules à poids lourds, étant donné l'échelle variable de charges qu'ils sont appelés à porter.

Toutefois, ces critères sont très difficiles à satisfaire si l'on ne se limite qu'à l'utilisation des composantes de suspensions du genre classique. Une suspension ayant une raideur et/ou un taux d'amortissement variables est la seule qui puisse rencontrer les besoins requis. En principe, c'est donc une suspension "active", et non passive, qui est nécessaire.

SUSPENSIONS "ACTIVES".

Indépendamment de la direction et de la vitesse du déplacement de la roue (ou de l'essieu, s'il y a lieu), une suspension "active" peut ou bien résister à une contrainte externe, se déplacer selon l'intensité de cette contrainte et selon les instructions de son système de contrôle, ou encore, se déplacer en exerçant sa propre force de déplacement. Par contre, l'ensemble d'un ressort et d'un amortisseur classique (une suspension passive) ou d'un amortisseur à taux variable (une suspension "semi-active") ne peut exercer une résistance qu'en fonction de la direction et de la vitesse de la roue, suite aux contraintes externes.

Une suspension "active" se distingue donc des autres types de suspensions du fait qu'elle soit capable de modifier sa raideur et son amortissement de façon quasi-instantanée selon la charge imposée, la condition de la chaussée et le comportement dynamique du véhicule.

Il existe plusieurs genres de suspensions adaptatives. Celles-ci peuvent maintenir une garde-au-sol constante par l'entremise d'un contrôle mécanique, ou peuvent varier leur amortissement selon une échelle limitée et prédéterminée de certains taux d'amortissement. Ces changements ne se font pas de façon instantanée et conséquemment ces suspensions ne sont pas du genre "actif".

AVANTAGES D'UNE SUSPENSION ACTIVE.

Certaines caractéristiques des suspensions "actives" font preuve d'avantages par rapport aux suspensions passives. Voici quelques unes de ces caractéristiques:

-Les "ressorts" et "amortisseurs" ont la capacité de se modifier en fonction de la charge imposée à chaque roue (au repos) et/ou en fonction d'une variation de charges susceptible de se produire en mouvement (consommation de carburant, accumulation de neige, etc.). Cette caractéristique permet de réduire ou d'éliminer l'affolement des essieux et la détérioration de la chaussée qui en est la conséquence.

-Indépendamment du placement des charges distinctes imposées sur le plateau de la remorque, les nivellements transversal et longitudinal de cette dernière sont contrôlés et ajustés de façon continue, au repos ainsi qu'en mouvement.

-La raideur et l'amortissement de chaque unité de suspension (vérin dans le cas d'une suspension "active" hydraulique) peuvent être ajustés en roulant, de façon instantanée et indépendante, afin de répondre aux déplacements et accélérations imposés par la route à chaque roue (ou essieu, selon le cas). Ces ajustements interviennent aussi dans le cas d'accélérations latérales, de pentes, etc., afin de résister aux retournements qui figurent dans 7% des accidents des véhicules articulés commerciaux.

-Selon les limites mécaniques de la suspension, la garde-au-sol d'un système actif est variable et peut être augmentée ou diminuée selon le besoin et le choix du conducteur. A basses vitesses, sur des surfaces accidentées, une plus grande garde-au-sol peut être utile pour éviter les accrochages. A hautes vitesses, sur parcours lisses, la garde-au-sol peut être diminuée, question de réduire le coefficient de traînée.

-Puisque la suspension est reliée à un système de contrôle électronique, un tableau de visualisation peut facilement y être intégré. Ce genre de tableau permet à l'opérateur de connaître la disposition des charges de sa remorque (camion, autocar, etc.), la charge imposée à chaque essieu (ou roue), la marge instantanée de parallélisme du véhicule par rapport au sol (longitudinalement et transversalement), de même que plusieurs autres renseignements. Un voyant lumineux et/ou un klaxon pourraient avertir l'opérateur d'un danger imminent, tel qu'une accélération latérale excessive. Cet avertissement pourrait permettre une réduction de vitesse ou autre manoeuvre rectificative avant que n'intervienne un retournement ou autre incident.

LA REDUCTION DE LA DETERIORATION DES CHAUSSEES.

Selon plusieurs études, la détérioration des chaussées s'exprime en termes d'une fonction proportionnelle à la quatrième puissance de la charge d'essieu. Donc, afin de réduire cette détérioration de moitié (50%), il suffit de diminuer par moins de 16% la charge dynamique de l'essieu. Une réduction de cet ordre, par rapport à un système de genre classique, a déjà été obtenue lors d'essais sur un train d'atterrissage "semi-actif".

Il s'ensuit que l'utilisation d'un système de suspension "actif" ou "semi-actif" dans le cas d'un véhicule à poids lourds effectuerait de semblables réductions au niveau des réactions dynamiques imposées aux chaussées et, conséquemment diminuerait le taux de détérioration de ces dernières. Un des résultats majeurs de l'utilisation générale de ce genre de suspensions dans le contexte des véhicules à poids lourds se rapporte aux importantes économies de maintenance des routes. D'après certains, la réduction de la maintenance des chaussées représente l'investissement le plus rentable au niveau du transport routier.

CONFIGURATIONS DES SUSPENSIONS ACTIVES:
VEHICULES A POIDS LOURDS.

Pour ce qui est des véhicules routiers à passagers, le sujet des suspensions "actives" a déjà été bien étudié. Il existe plusieurs prototypes perfectionnés qui roulent et démontrent les avantages offerts par de tels genres de suspensions. La configuration type qui s'est imposée dans ce domaine ressemble d'assez près aux suspensions hydrauliques pour véhicules à passagers qui existent depuis trente ans: un vérin hydraulique placé entre la masse suspendue et la masse non-suspendue (à chaque roue), alimenté par un circuit central en approvisionnement d'huile et dont les caractéristiques dynamiques sont déterminées par des soupapes mécaniques qui dirigent le flux hydraulique.

Dans le cas d'une suspension "active", les soupapes mécaniques sont remplacées par des servo-soupapes proportionnelles actionnées par des signaux électriques. Puisque les vérins ne sont plus raccordés à un ressort hydropneumatique, le flux continu (et réversible) de l'huile sert de ressort "synthétique". Des ressorts mécaniques sont aussi souvent présents afin de supporter la masse statique du véhicule et ainsi réduire l'énergie utilisée au niveau de la pompe hydraulique. Dans ce cas, les vérins servent à la fois de stabilisateurs dynamiques et de résistances d'appoint pour une charge variable (que cette charge soit statique ou dynamique).

En principe, ce même genre de configuration de suspension "active" peut être adopté par les véhicules à poids lourds, ainsi que par les systèmes de transport par voies ferrées. Toutefois, quelques changements sont nécesaires dans le cas de véhicules modulaires, tels que tracteurs et remorques. Cette qualité modulaire entraîne un échange d'unités, d'où la nécessité de s'assurer qu'un tracteur équipé d'une suspension "active" puisse s'adapter sans ennui à une remorque ayant une suspension classique. De la même façon, une remorque équipée d'une suspension "active" ou "semi-active" doit être en mesure de s'adapter à n'importe quel tracteur sans que le fonctionnement de sa suspension ne soit affecté.

La vérification de l'étanchéité et la purge d'une suspension hydraulique requiert un équipement spécialisé. Cette procédure serait néanmoins nécessaire à chaque adaptation du tracteur et de la remorque, si la suspension de cette dernière dépend d'un système à pouvoir hydraulique. De plus, une telle remorque ne serait compatible qu'avec certains tracteurs pourvus d'un réseau hydraulique compatible. Cette solution n'est guère pratique.

Afin qu'une remorque à suspension "active" ne soit pas dépendante d'un circuit hydraulique provenant du tracteur, il serait possible de l'équiper d'une pompe en approvisionnement d'huile reliée à un petit moteur. Ce moteur pourrait fonctionner en utilisant de l'air comprimé en provenance du tracteur, ou encore par combustion interne, comme dans le cas des remorques réfrigérées. Mais ces deux solutions pourraient s'avérer inefficaces, en raison de la consommation excessive d'air comprimé dans le premier cas, et des coûts d'installation, de maintenance, et d'opération dans le second.

L'utilisation directe du circuit d'air comprimé provenant du tracteur est possible mais ne permet pas à ce moment-là une suspension "active", puisque cet air est compressible. Cette caractéristique n'empêche pas toutefois l'adoption d'une suspension "semi-active". D'après certaines études, les caractéristiques dynamiques d'une telle suspension se rapprochent de très près de celles des systèmes complètement "actifs", plus particulièrement dans des circonstances où sont prévues une raideur variable (non-instantanée) ainsi qu'une plage généreuse du débattement des roues.

Pour ce qui est d'une suspension "semi-active", les vérins et servo-soupapes proportionnelles du système "actif" sont retenus et reliés à un accumulateur hydropneumatique. Ce dernier sert de ressort à raideur variable, selon la pression d'air qui s'y trouve. Cette raideur n'est pas variable de façon instantanée mais peut quand même être modifiée rapidement selon les changements de charges ou les accélérations perçus par le système de contrôle (temps de

réponse de l'ordre de 1 Hertz). D'autre part, la pompe en approvisionnement d'huile, ainsi que les circuits et accumulateurs hydrauliques du système "actif" peuvent être supprimés, permettant de ce fait des économies énergétiques.

Cette suspension "semi-active" retient donc la capacité de faire des changements du taux d'amortissement instantané, ainsi que d'adopter une raideur idéale selon la charge imposée. Une remorque équipée de ce genre de suspension serait compatible avec un tracteur du type standard, puisque ce dernier doit tout de même servir comme source de pouvoir électrique et d'air comprimé pour le système de freinage, sinon pour la suspension de sa remorque. Par contre, ce tracteur pourrait lui aussi être équipé d'une suspension "semi-active" ou "active" avec son propre système de contrôle.

En résumé, la configuration proposée pour les véhicules à poids lourds, au niveau des suspensions du genre "actif", est la suivante:

TRACTEUR ou AUTOCAR: Suspension "active" ou "semi-active", à base de vérins hydrauliques et servo-soupapes proportionnelles. Système de contrôle distinct au tracteur. Ressorts de charge statique du type traditionnel ou selon la capacité des accumulateurs hydropneumatiques, dans le cas de la suspension "semi-active".

REMORQUE: Système "semi-actif", à base de vérins hydrauliques, servo-soupapes proportionnelles et accumulateurs hydropneumatiques. Système de contrôle distinct à la remorque. Possibilité de ressorts de charge statique du type traditionnel, selon la capacité des accumulateurs hydropneumatiques.

CONCLUSIONS ET RECOMMANDATIONS:

Les suspensions "actives" et "semi-actives" peuvent offrir plusieurs avantages pour les véhicules à poids lourds: parmi les plus importants sont la réduction des réactions dynamiques imposées aux chaussées et une augmentation de la sécurité d'opération de ces véhicules.

Un prototype d'une suspension "active" installé sur un véhicule à passagers a d'ailleurs déjà fait ses preuves sur des chaussées accidentées. Une étude expérimentale du même genre ayant comme objectif une démonstration d'une suspension "semi-active" pour les véhicules à poids lourds devrait être entreprise.

TABLE OF CONTENTS

1. INTRODUCTION 1

2. TYPES OF HEAVY VEHICLE SUSPENSIONS 3
   2.1 Dynamics Classification Overview 4
   2.2 Nominal Suspension Characteristics 6

3. ACTIVE SUSPENSION CRITERIA 9
   3.1 Environmental Criteria 9
   3.2 Mechanical Criteria 10
   3.3 Sensing and Control Considerations 12

4. HEAVY VEHICLES AND ACCIDENTS 15
   4.1 Heavy Vehicle Accident Types and Distribution 15
   4.2 Accidents and Suspension Dynamics 21

5. HEAVY VEHICLE OPERATING COSTS 25
   5.1 Infrastructural Costs 25
   5.2 General Vehicle Operating Costs 27
   5.3 Active Suspensions and Operating Costs 28

6. ROAD WEAR AND DAMAGE 31
   6.1 Road Damage Types 31
   6.2 Road Damage Caused by Suspension Dynamics 32
   6.3 Active Suspensions and Road Damage 36

7. HARDWARE REVIEW 39
   7.1 Existing Hydraulic Equipment 39
   7.2 Existing Electronic Equipment 41

8. ACTIVE SUSPENSION DESIGN 43
   8.1 Active Suspension: Mechanical Configuration and its Governing Parameters 43
   8.2 Active Suspension: Electrical Configuration and its Governing Parameters 46
   8.3 Mechanical Parts Specification 50
   8.4 Electrical Parts Specification 51
   8.5 Active Suspension: Software Configuration and its Governing Parameters 52

9. ACTIVE SUSPENSION DYNAMICS 57
   9.1 Identification of Safe Dynamic Behaviour, and the Effect of Active Suspensions 57
   9.2 Rolling Resistance Sources and the Effect of Active Suspensions 58
   9.3 Advantages of an Active Suspension 59

10. CONCLUSIONS AND RECOMMENDATIONS 61

11. REFERENCES 63

APPENDIX A - Hydraulic Suspensions

TABLE OF ILLUSTRATIONS

ILLUSTRATION		PAGE
1.	COMPARISON OF SUSPENSION TYPES	7
2.	AIR AND HYDRAULIC SUSPENSIONS	8
3.	PASSIVE, SEMI-ACTIVE AND ACTIVE MODELS	48
4.	EXAMPLE OF A FORCE GENERATOR	49
5.	VALVE	54
6.	EXAMPLE OF AN ACTIVE SUSPENSION FOR AN ARTICULATED HEAVY VEHICLE	55

1. INTRODUCTION

The causes and mechanisms of road wear have been the subject of considerable international research, particularly of late. Much of this attention is now being directed towards the contribution to pavement deterioration made by commercial vehicles.

Some studies have identified certain types of heavy vehicle suspension systems as being potentially more damaging than others, particularly in terms of their reactions to certain types of road surface irregularities or vehicle dynamics. Although not as fully researched, the influence of such reactions on the quality of the vehicle's operating stability and safety can be considered to be substantial and to vary according to the suspension's design characteristics.

A general review of heavy vehicle suspension systems would allow a qualitative identification of characteristics which contribute to the undesirable reactions described above. This review could then serve as a frame of reference for the evaluation of the potential level of usefulness of an active suspension (an evaluation of this type is one of the main objectives of this program). A high level of usefulness would be present if the implementation of an active suspension resulted in economies through reduced road wear and a smaller number of road accidents. These savings would form the basis of a justification for the use of such a system, as well as determine whether it would be considered a necessity in a given operating context.

In a 1978 ASME paper, D.C. Karnop asked the question: Are Active Suspensions Really Necessary? (59). He answered his own question in this manner: "Using linear theory and traditional frequency response plots, it is shown that in the most basic case, a worthwhile, but not crucial improvement can be gained by the use of active or semi-active systems, while in a more complex case the best possible passive suspension performance is very poor when compared with that achievable using an active system. ...even a fairly simple type of active force supplying part of the suspension force can drastically improve performance. In such cases, the active suspension can be the key to the success of the entire system." (59).

An active suspension may be the key to greater safety and general economy of operation for both heavy vehicles and road infrastructures. The present study will investigate this subject and render conclusions regarding the practical use of such suspension systems as well as estimates of the possible reductions in accidents and road wear.

2. TYPES OF HEAVY VEHICLE SUSPENSIONS

Since their inception at the turn of the century, road vehicles of various types have evolved for various purposes. Passenger cars, for example, have specialized in the transportation of small numbers of people. As their primary purpose has been that of carrying a relatively sensitive cargo, their suspension systems have been designed to offer a good ride. While serving the same basic purpose, buses are required to operate over a wider range of laden-to-unladen mass ratios. Despite this disadvantage, their suspension systems must offer a ride comparable to that of passenger cars, while retaining adequate handling and control characteristics.

Trucks, for the most part, have been designed to carry dead loads. These loads can be of the order of several times their own weight. As the laden-to-unladen mass ratio of the truck or tractor-trailer varies, only small variations in ride height can be accepted. Unless compensation mechanisms are incorporated into the suspension, this requirement is usually translated into a system having relatively high spring rates and small displacements.

A large number of different types of suspension systems have been introduced in order to cope with the varied service loads involved in the operation of heavy vehicles. The reactions introduced into the pavement by these suspensions are dissimilar and depend to a great extent on their mechanical configuration and their dynamic characteristics. As the perceived safety of the vehicle is directly related to the level of stability of its suspension, this last characteristic is a most important one and must not be neglected when selecting a system for a particular application.

Applied research into the relative safety and pavement-reaction levels of various types of suspensions has been performed in a number of countries, including Canada. Although most of this work has concentrated on the suspension at the rear of tractors and on the linked axles of trailers, some studies have also investigated various types of suspensions used on the front axles of tractors.

The following sections list the most common suspension types for front and rear axles, as well as an evaluation of their dynamic characteristics, according to published research findings. A relative scale with ratings of high, medium and low are used to grade the road impact and stability values of each suspension type. Quantitative values have not been used in these lists as they are not usually available from studies (referred to by the number in brackets) which tend to compare suspensions in relative, rather than absolute, terms.

2.1 Dynamics Classification Overview

SUSPENSION TYPE	DYNAMICS EVALUATION HIGH *** MEDIUM ** LOW * ROAD IMPACT	STABILITY
-Single Front Axle Type:		
1. Steel leaf spring (common). -front fixed, rear slipper -front fixed, rear shackle -front slipper, rear fixed -front shackle, rear fixed -shackle at both ends, with radius rod -slipper at both ends, with radius rod -any of the above with a rubber or urethane overload/auxiliary spring (such as AEON or other). -any of the above in single stage, two-stage progressive, main and helper, variable rate, variable rate two-stage, progressive. -any of the above with or without (under 5,500 Kilos) auxiliary damping. -any of the above with a rebound leaf.	**	*(23)
2. Air springs (buses primarily). -single air spring with damper. -multiple air springs with damper. -air spring(s) with air pressure sensitive, variable rate damper.	*(33)	***
-Double Front Axle Type:		
-steel leaf spring (most common). -dual configuration of a single front axle suspension type.		

N.B. Illustrations 1 and 2 (on pages 7 and 8) depict figurative views of certain suspension types.

SUSPENSION TYPE -Rear Axle:	DYNAMICS EVALUATION HIGH *** MEDIUM ** LOW * ROAD IMPACT	 STABILITY
-Single Rear Axle Type:		
1. Steel leaf spring (common).		
-front fixed, rear slipper	**	*(23)
-front fixed, rear shackle		
-variable rate, slipper ends contacting cam type brackets	**	*(23)
-shackle or slipper at front and rear, with radius rod(s)	**	*(23)
2. Air springs (less common).		
-single air spring with damper, on rigid trailing arm.	*(33)	***(82)
-single air spring with damper on leaf spring type trailing arm.	*(33)	***
3. Rubber springs (common in Europe).		
-pads in compression, no damper.		
-pads in shear, no damper.		
-hollow variable rate donuts, in compression, without damper.	***(52)	***(52)
4. Torsion bar spring (less common).		
-longitudinal torsion bar with crankarm link, with or without damper.		
-Double Rear Axle Type:		
1. Steel leaf spring (most common).	***(19)	**
-walking beam type	***(12)	**(90)
-four spring type	**(12)	***(90)
-two spring type	***(48)	**(48)
-underslung spring		
-overslung spring	***(52)	**(52)
-cantilever spring type	***(21)	*
2. Air springs (common)		
-leading/trailing arms both with dampers	*(48)	***(48)
-walking beam type	*(48)	***(48)
3. Rubber springs (common in Europe).		
-pads in compression, no damper.		
-pads in shear, no damper.		
-donuts in shear, no damper.		
-donuts in torsion, with damper.	***(57)	**(57)
-hollow variable rate donuts, in compression, without damper.	***(52)	**(52)
4. Torsion bar spring (less common)		
-longitudinal torsion bar with dual crankarm links, with or without dampers.	*(12)	*(90)
5. No springing (common off-road)		
-walking beam type.	***	**
-Auxiliary Axles:		
-air springs (most common).		

2.2 Nominal Suspension Characteristics

The following are typical values for passive suspensions:

FRONT AXLE: Nominal values.

Load Range -unloaded: 45 kN (10,000 pounds)
-loaded: 90 kN (20,000 pounds)
Nominal loaded-to-unloaded ratio: 2 to 1

Spring Rate Range, Nominal: 105 N/mm to 525 N/mm
(600 lbs/in to 3000 lbs/in)

Static Deflection Range: 90 mm to 130 mm, lower rates
60 mm to 100 mm, higher rates
(3.5 in. to 5 in., lower rates
2.5 in. to 4 in., higher rates)

Specific Frequency Range: 2 to 1.3 Hertz (or higher, depending on the level of hysteresis or friction present in the suspension)

Damping Ratio: An effective damping ratio is difficult to establish in the case where dampers are not present as such. A nominal friction ratio for a heavy-vehicle suspension could be given as .02, yielding nominal resonant frequencies in the area of 2 Hertz. Where dampers are provided, a nominal damping ratio range is from .2 to .4.

REAR AXLE: Nominal values.

Load Range -unloaded: 53 kN (12,000 pounds)
-loaded: 102 kN (23,000 pounds)
Nominal loaded-to-unloaded ratio: 2 to 1

Spring Rate Range, Nominal: 400 N/mm to 1500 N/mm
(2300 lbs/in to 6000 lbs/in)

Static Deflection Range: 50 mm to 130 mm
(2 in. rubber or steel spring, up to 5 in. for air springs)

Specific Frequency Range: 1 to 4.4 Hertz theoretical, affected by friction and hysteresis.

Damping Ratio: As above, for the front axle case.

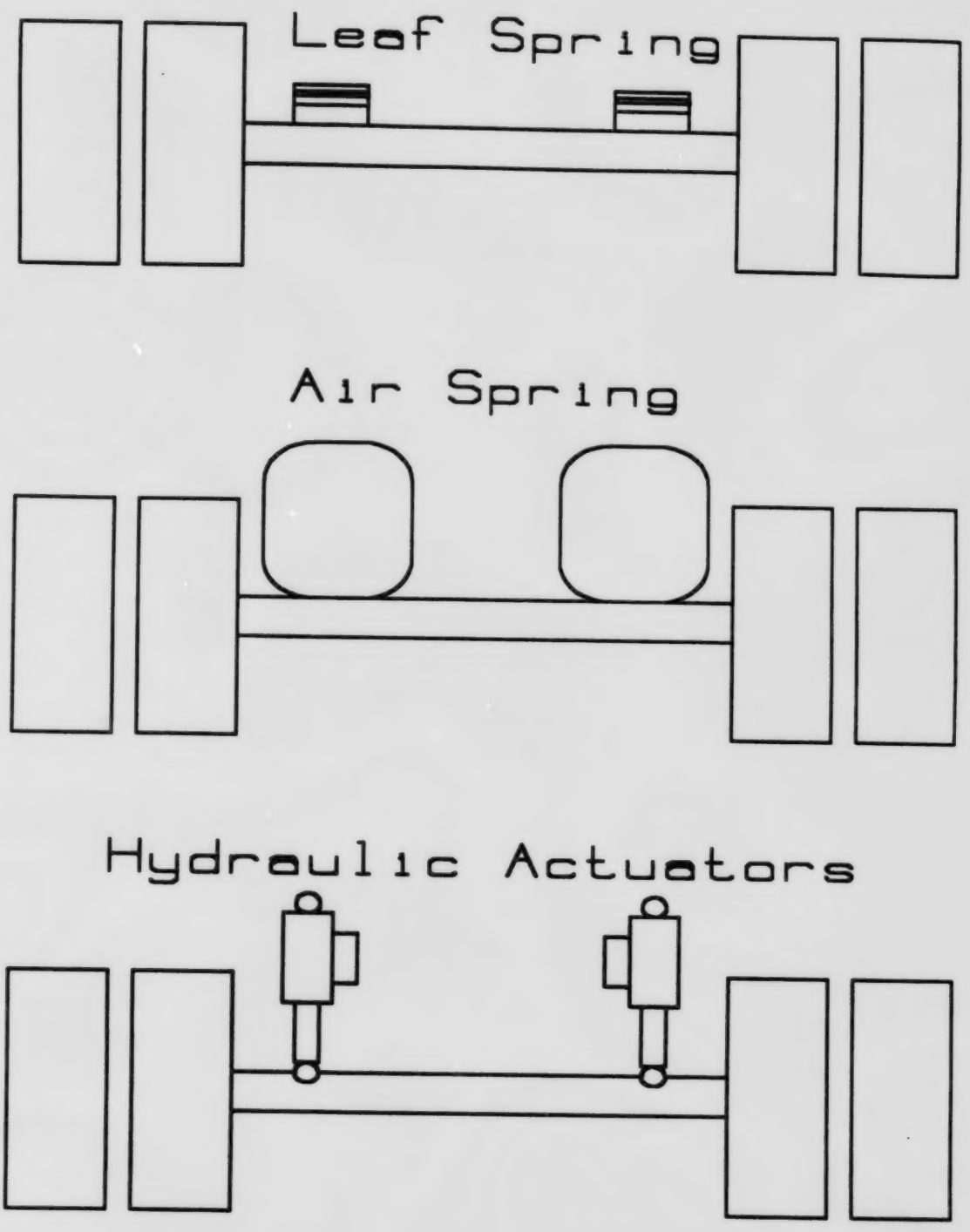

ILLUSTRATION 1: Comparison of suspension types.

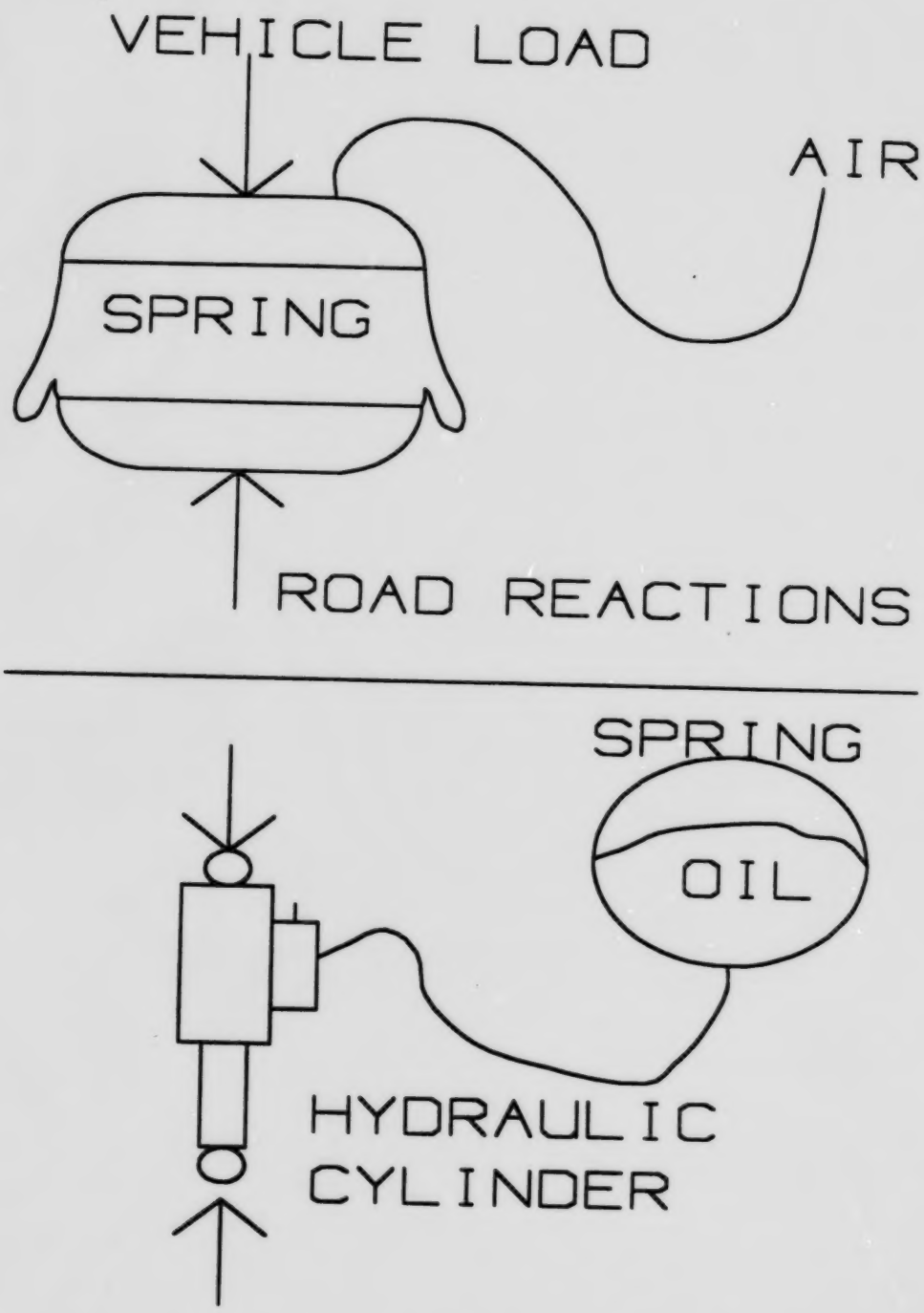

ILLUSTRATION 2: AIR AND HYDRAULIC SUSPENSIONS

3. ACTIVE SUSPENSION CRITERIA

Commonly used heavy vehicle suspensions are designed in such a way as to attenuate dynamic loadings. The severity of these loadings depends upon the characteristics of both the travelled surface and of the vehicle, as well as the velocity at which the vehicle passes over this surface.

3.1 Environmental Criteria

From the point of view of road roughness, an ideal vehicle suspension should have as low a natural frequency as possible, relative to the excitation frequency imposed by the road. A value of 1 Hertz, for example, might be considered quite good for a current heavy vehicle suspension. A natural frequency of this order implies large static deflections, or, springs of relatively low rates. Certain air suspensions, having an axle travel space of approximately 120 mm (up or down, respectively), can approach these frequency values.

In terms of present-day suspension equipment, a low natural frequency is usually translated or interpreted as representing 'weak springs'. Such low rate springs are desirable in certain circumstances, such as when the suspended vehicle's inertia can be exploited to maintain stability when it is subjected to high frequency excitations.

Unfortunately, the inertia, or mass, of a heavy vehicle does not remain constant. Springs of a given rate designed for use at a certain mass level are not ideal at another mass level. If the mass becomes too great in relation to the spring's rate, under excitation the suspension will tend to meet or exceed its intended deflection limits. Excessive dynamic reactions ensue, reactions which must be absorbed by the vehicle's structure, its suspension, and the source of the excitations: the road.

Conversely, if the mass is too small in relation to the spring's rate, road excitations will have little effect on spring deflections. Thus, the suspension and the vehicle will essentially act as a solid body, reacting directly to the accelerations imposed by these excitations. Here again, these reactions must be absorbed by the vehicle and the road, resulting in structural fatigue or excessive wear to both.

In addition to coping with varying vehicle mass and road accelerations, the suspension must maintain the vehicle's stability in situations where it is subject to lateral accelerations, wind-loadings, and other forces. The stiffness of the springs in the suspension, for example, is of considerable importance in determining the roll-over

threshold of a given heavy vehicle. This threshold is commonly used as an indicator of the vehicle's level of stability. A suspension design having high spring rates usually exhibits a high rolling resistance, while a low spring rate suspension is usually more subject to rolling behaviour (93). It should be noted that several other factors affect the actual roll characteristics of the entire vehicle.

From a mechanical point of view, the performance requirements of a typical heavy vehicle's suspension system can thus be summarized as
- -low natural frequency
- -low dynamic reactions at the deflection limits
- -variable mass range
- -high rolling resistance
- -high stability.

From an electrical or electronic point of view, the performance requirements of a typical heavy vehicle's suspension system are less clear. Little use of non-mechanical technology is used in such systems. One exception is in certain air-suspensions, where the ride height may be adjusted by means of electrically actuated solenoid valves. The required response time of such valves is quite low, on the order of 1 Hz. It is thus reasonable to propose that currently-used heavy vehicle suspension systems have little or no identifiable electrical/electronic performance requirements. However, any electronic equipment intended for use on heavy vehicles should be designed according to the Society of Automotive Engineers (S.A.E.) standard J 1455 - Recommended Environmental Practices for Electronic Equipment Design (Heavy Duty Trucks).

3.2 Mechanical Criteria

An active suspension intended for use on heavy vehicles must behave at least as well as current suspensions do. For example, it would have to equal or surpass the level of stability of a walking-beam type of suspension, while maintaining its wheel-to-road reactions at an equivalent or lower level than those of an air-type suspension. As well, it would be desirable to have its roll characteristics approach or surpass those of an equivalent non-elastic system, in order to reduce the incidence of roll-overs.

While attempting to emulate the above generally described behaviour, an active suspension would be subject to the following environmental inputs:

-Road Excitation Frequencies: A typical frequency range of road acceleration inputs is from 0 to 30 Hz (79). Unpaved roads such as those used for the recovery of natural

resources (mining and forestry), or, in the construction of remote infrastructures such as hydroelectric dams may exhibit higher acceleration input frequencies (up to or exceeding 40 Hz). The undesirable wheel-hop mode of passive suspensions is usually initiated in the 10 to 15 Hz range (1), which is within normal operating conditions.

-Road Excitation Amplitudes: In those cases where the accelerations imposed by typical road excitations range from 5 to 25 G's (94), corresponding displacements of approximately 13 centimetres can be reached (94). Unpaved or extremely rough roads can induce higher amplitudes than these. An active suspension designed for use on such roads would have to deal with commensurately higher accelerations and displacements.

-Lateral Accelerations: Lateral acceleration rollover thresholds of typical heavy vehicles lie in the general range of .2 to .35 G's (69). An active suspension would be required to allow the vehicle to develop an equivalent or greater level of lateral acceleration without rolling over.

The basic mechanical characteristics of an active system designed to deal with the above inputs would be similar to the following:

-Characteristic Frequency: 1 Hz or less, initial effective rate, under all loading conditions (79). A spring rate which is as low or lower than the natural frequency of the vehicle helps in reducing the level of fatigue-inducing stresses to which the latter is subjected. The characteristic frequency of the suspension in a classical system is the natural or resonant frequency of the effective spring and damper assembly (including friction). In an active system, an initial effective rate of this order would be emulated by an appropriate modulation of the artificial springs' force/displacement generators. A hydraulic artificial spring could do this by means of a metered fluid flow, controlled according to the dynamic conditions imposed upon the 'sprung' mass.

-Variable Effective Frequency: From less than 1 Hz to that of the locked suspension (essentially static). The initial adoption of a low effective rate presages, rather than precludes, the implementation of alternate effective rates or force values. These alternate values would be used in order to deal with inputs of various excitation frequencies and amplitudes, as well as combined inputs such as simultaneous lateral and vertical accelerations.

On an approach ramp, for example, a vehicle's suspension system would be required to support the sprung mass while resisting lateral accelerations and weight transfers, as well as complying with any bumps or depressions which might

be encountered. The required range and resolution of effective frequencies in a true active system would thus be quite vast. Such a system's force/displacement generators would alternately be called upon, in reference to a given baseline position, to retract, remain static or extend. The velocity of retraction or extension could be compared to the frequency or rate of a traditional spring and damper assembly. A very rapid actuation would correspond to a suspension of very low stiffness and damping: one having a low characteristic frequency (less than 1 Hz or so). In contrast, a rigid or static artificial spring would be likened to a suspension of very high stiffness and damping: one having a very high characteristic frequency.

-Mechanical Response Time: From 250 Hz to 500 Hz (phase lag of approximately 90 degrees). In the case of a hydraulically actuated active suspension, this signifies that the control valves metering the fluid flow to and from the force generators would be required to switch 'on' or 'off' (open and close) at least 250 times per second. In order to allow year round operation of a so-equipped vehicle, the response time of the valves would have to be unaffected by changes in ambient temperature.

Experimental results from the installation of a prototype active suspension in a passenger car have demonstrated that a mechanical response frequency of 250 Hz is an acceptable lower limit (32, 125). A response frequency lower than this permits a greater amplitude transfer of excitation inputs and results in subjectively lower ride and stability characteristics. As the typical heavy vehicle tire is less compliant than its passenger vehicle equivalent, a higher mechanical response frequency may be required for the safe use of an active suspension on heavy vehicles (70).

3.3 Sensing and Control Considerations

Assuming that physical components can be designed in such a way as to comply with the above requirements, the electronic sensing and control system essential for the proper management of these components must itself be capable of meeting certain performance criteria. The control system must be capable of sensing and recognizing what is happening at each suspension location, as well as to the vehicle as a whole, then calculate and implement the optimal response. As the dynamic events being monitored and controlled in such a situation occur at a relatively high frequency, the response time of the system itself must be quite low.

-Sensing Capabilities: Wheel load, wheel position (point in suspension travel), rotational wheel velocity, vehicle speed, vehicle attitude and vehicle acceleration(s) are the main external values to be measured by sensors (45).

-Sensing Ranges: A minimum sensing resolution of the order of 1 part in one thousand parts is expected to be required. In terms of wheel load, for example, the sensitivity of the given sensor should be such that a change in overall load of no less than .1% would be noticeable, at the wheel in question.

-Sensing Rates: If 250 Hz is considered the lower limit in mechanical response time, the vehicle data measured by the sensors should be renewed and sent to the control system at, ideally, a much greater rate (at least twice as great). As the sensing rate increases beyond the basic mechanical response time, a greater number of control strategies are available to the designer of the control system, allowing better management of the suspension's possible reactions. The practical lower limit in sensing time is limited by the control system's maximum sampling frequency, which is a function of the control algorithm and of the host microprocessor or microcontroller.

-Data Processing Rate: The control system's processing speed must be high enough to permit several sensing and calculation cycles between each mechanical actuation instruction. Too low a rate would not allow the control system to determine the 'trend' or slope of the dynamic event at hand, and would thus hamper the selection or specification of an optimal actuator setting. The actual processing speed depends on the control algorithm as well as on the architecture of the control system. The lower bound or minimal processing speed criteria will be calculated and specified as a function of the vehicle's intended top speed, its lateral and longitudinal acceleration performance requirements, the roughness of the surface it is designed to run on, the response time of the mechanical components, the transmission delay due to data link lengths and an associated safety factor.

-Signal Type: The use of analog servo-equipment in conjunction with a digital control system can induce time delays due to the conversion of digital commands into analog signals (for most control purposes the reverse conversion, Analog to Digital, can essentially be considered as an instantaneous one). The specification of servo-equipment capable of directly utilizing a digital signal, where possible, would permit a faster overall response time, as well as reduce the complexity of the control system.

-Data Transmission Distances: The transmission of data at high rates can induce signal reflectance, a source of errors. High characteristic impedance over long transmission line distances (more than a few metres), mismatched carrier-to-receiver or carrier-to-sender impedances as well as the actual data transmission speed all influence the amount of

reflectance which is generated. A tradeoff thus exists between the maximum length and quality of the transmission line and the permissible speed of data transmission on that line. On a heavy vehicle, the distances between the location of the axles (sensors and actuators) and the potential position of an active control system are thus non-negligible and likely to influence the type of transmission line adopted, the data transmission speed and/or the location of the controller.

-Environmental Factors: All of the electronic or electrical equipment associated with the active suspension must be capable of functioning under the ambient climatic conditions to which any other suspension system is typically subjected. The equipment must also be packaged and/or mounted in such a way as to resist damage from vibrations and shock loadings.

4. HEAVY VEHICLES AND ACCIDENTS

Motor vehicle accidents are costly events. Such accidents cause the loss of equipment, time, and lives. Insurance industry statistics indicate that the average compensation cost for an injury resulting from a commercial vehicle accident is four times as great as the average of all other types (76). Further, once a heavy vehicle has been involved in an accident, it is twice as likely to be a fatal one, in comparison to those involving only passenger vehicles (76). A considerable social gain can thus be obtained through the reduction or elimination of heavy vehicle accidents.

The safety and crashworthiness characteristics of heavy vehicles are now under close scrutiny in North America. An act of the American Congress (Motor Carrier Safety Act of 1984) has called for the preparation of a report dealing with the subject of truck safety (that report, at the time of this writing, is not yet finished). In Canada, an unprecedented vehicle weights and dimensions study has recently been completed, dealing with many aspects of heavy vehicle stability and safety (49).

In a historical context, these actions form part of the effort initiated in the late 1950's towards an improved level of road vehicle safety through the generation of legislated standards concerning the design, maintenance and use of such vehicles. However, the recent deregulation of the transportation industry in the United States and the prospect of a similar deregulation in Canada have precipitated concerns over the safety of operation of heavy vehicles in an increasingly competitive market. This has encouraged a closer examination of the conditions surrounding and contributing to the incidence of heavy vehicle accidents.

4.1 Heavy Vehicle Accident Types and Distribution

Many elements may be involved in, or contribute to, the occurrence of a heavy vehicle accident. The classification of such an accident can thus be made according to those factors identified as having been associated with it. Four main factors have repeatedly been cited and will be treated separately: Operator, Vehicle, Environment and Cargo (76).

OPERATOR: A large number of accidents are blamed on driver error. In a survey of commercial vehicle accidents during a time period spanning from 1979 to 1981, the Ontario Ministry of Transportation and Communications reported that "88% (417) were attributed to the actions of the driver of the vehicle precipitating the accident" (41, p. ix). It should be noted, however, that the "vehicle precipitating the

accident" is not always a commercial one. This same study indicated that the commercial articulated vehicle (CAV) driver was at fault in 43.8% of the "total number of multiple vehicle accidents reported", whereas an automobile driver was found to be at fault in 56.2% of these cases.

A study prepared by the Traffic Safety Engineering Branch of the Saskatchewan Highways and Transportation observed that "in the multi-vehicle truck accidents, inattention by the other vehicle operator", not the truck driver, "was 3 times more prevalent" (42, p. 26). However, 'inattention' on the part of the CAV driver was described as a main contributing factor in accidents involving single heavy trucks.

In reaction to this type of information and statistics, R.V. Durham, Director of Safety and Health for the International Brotherhood of Teamsters, has stated that "Until the carriers, and the drivers, decide there is a better approach, we will continue to see statistics that suggest the truck driver is responsible for 94% of the accidents" (76, page 49). Although stated defensively, this comment on the part of a union representative makes it seem obvious that driver error or mis-judgement is acknowledged to be an important contributing factor in the occurrence of commercial vehicle accidents. If this is the case, what can be done to minimize or reduce this contribution?

Effective operator training and the enforcement of safe driving schedules are two means by which the level of driver-induced accidents may be reduced. A lack of familiarity with a given vehicle or an insufficient understanding of vehicle stability in general have been noted as present in cases where the CAV driver was determined to be at fault (41, p. ix). For example, in a review of certain types of accidents, Richard Radlinski, of the NHTSA, observed for a specific case that "the driver was quite young, and made the comment that he had stopped the vehicle before with a load on it and never had any problem. He just did not understand why it jackknifed this time." (76, p.22). In relation to safe driving schedules, D.R. Linklater, of the New South Wales Department of Motor Transport, has noted that if truck operators "were to drive for less than 55 hours per week, their usage of alerting drugs and driving exposure (and concomitant driving fatigue) would be likely to be reduced below the critical level at which crash probability is increased" (135). It should be obvious that there is little that can be done, in terms of the mechanical equipment presently available, to reduce the accident exposure level related to poor operator training or to driver inattention and/or fatigue.

<u>VEHICLE</u>: An accident factor which originates from the vehicle itself is usually of one or the other of two major

types: the first is related to the design of the vehicle and its equipment, while the second can be traced to an inadequate maintenance program. Mechanical failure, for example, was cited as a cause of 12% of the commercial articulated vehicle (CAV) accidents over the course of a survey performed by the Ontario Ministry of Transportation and Communications (41, p. ix). Henry E. Seiff of the Motor Vehicle Manufacturers Association found that 9% of the fatal accidents involving CAVs were linked to vehicle-related contributing factors (80, p.5). No information concerning the actual nature or distribution of these mechanical failures was available from the above studies, i.e. whether the cited failures were blamed on a poorly designed component, an under-maintained one or an overloaded one.

The overall mechanical specification of a commercial vehicle and of its constituent parts (such as the suspension type and characteristics, brakes, etc.) determines the envelope of performance to which it is ultimately limited. In some cases, the design criteria used to determine the configuration of the vehicle, and of its components, does not necessarily address every applicable performance requirement. Suspension spring rates, for example, are determined primarily according to the ride rate and height required at full load, rather than according to the dynamic behaviour or the level of the vehicle's stability at various levels of load. It may thus occur that the actual design specifications and subsequent design-induced responses may encourage the incidence of CAV accidents. Sweatman and Lai, for example, have determined that the suspension type and rate, the vehicle's physical dimensions, and, other design factors, such as uninhibited liquid movement in tankers, can contribute to rollovers, which occur in "4 to 15 per cent of heavy vehicle accidents" (90, p.1).

Discrete part or component design, especially in terms of resistance to fatigue or deformation, can influence vehicle safety in those cases where an under-designed or poorly manufactured part fails (whether the failure is due to the part's physical configuration or its constituent material). In order to reduce the likelihood of such failures, Ray W. Murphy, of the Freightliner Corp., has found that the "measurement of load histories, theoretical durability analyses, programmed laboratory fatigue tests, and proving ground tests must all be incorporated" in the design and development of new heavy-duty vehicles (94, p.27).

A low level of maintenance can also lead to accidents classified as having been caused by vehicle factors. In a 1982 survey performed by the Bureau of Motor Carrier Safety (BMCS) in the USA, "33,174 vehicles were inspected and 32,510 had violations of the Federal Motor Carrier Safety Regulations (FMSCR). There were 12,564 violations discovered which resulted in a vehicle being placed out of service (38%

of the vehicles)." (68, p.17). Such violations, which are the result of an inadequate maintenance program, can contribute to or cause numerous types of accidents: "On steep long downgrades improperly adjusted brakes place higher demands on functioning brakes. When the pavement is wet improperly adjusted brakes can result in uneven braking that will cause a truck combination to jackknife" (68, p.17). On occasion, incompetent or negligent mechanics can effect 'repairs' which result in a vehicle becoming more prone to accidents. Corben (134) found a large number of such instances, one of which involved a fractured weld repair on a turntable (fifth-wheel) which was clearly marked 'DO NOT WELD': the result was an accident caused by the separation of the trailer away from the prime mover.

ENVIRONMENT:

In a recent study of truck control problems, R.D. Ervin, C.C. MacAdam and M.A. Barnes stated that the "accident file of the Bureau of Motor Carrier Safety (BMCS) for 1980 shows that 9% of all jackknife accidents and 16.8% of all truck rollovers occur on ramps. Clearly, such percentages are much higher than the fraction of total highway miles represented by ramp sections." (69, p.29). A number of environmental factors were identified as having contributed to the relatively high numbers of ramp accidents. These included excessive side friction factors (as compared to the roll stability limits of certain trucks), varying turn radii among the distinct curves comprising a given ramp, insufficient deceleration lane lengths prior to the ramp, poor drainage and/or pavement texture, curbs present at the outside of the ramp helping to induce overturning motions, and, as well, tight curves at the end of a downgrade ramp.

A similar environmental feature combined with other factors resulted in an accident described by Radlinski: "there was a 2% crown on the road. When the tractor drive wheels locked up, they rode down the crown causing the tractor to rotate." (76, p.22). In this case, it was determined that the poor condition of the brakes on both the tractor and the trailer, as well as a light load in the latter, contributed to the jackknife situation induced by the road. Road features such as railway grade crossings, in combination with certain vehicles, can cause specific types of accidents: "..lowboys, 'Nu-Car' Carriers, or house trailers ...may become hung-up due to the profile at the crossings. This type of accident appears to be occurring more frequently as roads and rails are raised during periodic maintenance and wheelbases are lengthened on specialty-type trailers." (68, p.18).

Lawrence E. Jackson, of the National Transportation Safety Board of the USA, examined two cases where the profile of the road surface were believed to have caused CAV accidents.

The first involved "a 1-to 1.25 inch depression which had been dug, replaced and sunk. When the truck rode over the depression the tractor's tandem equalizer beam failed and the truck overturned..." (68, p.21). In the second, "a truck broke its right bogie leaf spring assembly about a mile or two after running over a rough section of pavement. This pavement may have helped to strain the spring to such an extent that any minor pavement irregularity could have resulted in the fracture of the spring." (68, p.21). This truck eventually overturned, due to the broken bogie spring.

A further comment made in Jackson's report outlines the effect of certain types of road surfaces on heavy vehicles. It is quoted from a report on 'The Influence of Roadway Surface Discontinuities on Safety - A State-of-the-Art Report' which was published by the TRB in 1984: "Perhaps the one area of possible influence that has not been well addressed in the literature is the significance of special wavelengths of road roughness to which trucks may be sensitive. It is known among experienced truckdrivers that certain long wave undulations, as typified by pavement settlements in bridge approach areas, may be peculiarly difficult to negogiate with commercial vehicles, particularly tractor-semitrailers. These features tune to the low-frequency rigid-body bounce and pitch modes of these vehicles. Because the drivers are located near the extremities of the vehicle (far from the center of gravity), large displacement vertical and fore-aft motions can be imposed on the driver, thus complicating the task of maintaining control when negotiating these road features." (68, p.21). An example of this phenomenon was recorded in a test performed in 1985: "The testing showed that small bumps at 16-foot intervals could cause a bus to lose control. A large 1-inch mogul bump caused the sister bus to fishtail during a lane change." (68, p22).

Various other types of environmental factors can contribute to CAV accidents, such as the presence of snow and ice on the roadway, heavy rainfall or fog, poor lighting conditions, etc. In many of these cases the environmental factors are combined with certain driver related factors, such as excessive speed, and the result is an accident.

CARGO:

Cargo can be considered to be a contributing factor to CAV accidents in a number of ways. Probably the most obvious is the situation where a poorly anchored load becomes dislodged from its resting position and shifts or falls off the trailer. Numerous accounts of such accident causes have been documented, such as the following one by Corben: "On a road in a residential area a semi-trailer rolled over as it negotiated a right hand bend. The driver felt the load shift

and heard a loud bang before rollover. The load of paper rolls did not separate from the trailer. The load was contained in a frame of steel on the trailer with gates and tarps. The turn table was torn from the prime mover as the trailer rolled over. The trailer had been loaded higher on one side and in combination with vehicle speed on the bend had precipitated fracture of the turntable." (134, p.22). Corben goes on to state that "Load retention was noted to be a problem in causing crashes or injury by contributing to vehicle instability." (134, p.23)

The type and the initial position of the load can affect the stability and the dynamic characteristics of a heavy vehicle. Movements of liquid or non-fixed cargoes can contribute to unsafe situations: "...tank trucks may experience liquid surge and trucks such as those carrying meat may have swinging meat which add additional unstable forces that tend to turn vehicles over." (68, p.17). The same conditions were observed by Corben in Australia, where "Inadequate securing was noted for loads which were gas, fluid, semi-solid or large and unwieldy." (134, p.25).

In the case of tankers, for example, the vehicle's stability may be affected by the internal design of the tank as well as the amount of liquid inside it: "A disk baffle configuration... ...tends to maintain a much more stable characteristic at all liquid levels..." (65, p. 420). In this experiment, other types of baffles induced variable damping effects as the fluid level changed, resulting in an unpredictable dynamic load distribution on the vehicle.

Loading configurations of fixed cargoes can affect braking and traction characteristics of heavy vehicles. As truck tires are designed for low wear rates and manufactured of high-durometer rubber, axle loads have a direct influence over the level of traction which they can develop: "...loading does not appear to affect the passenger car but does lengthen the panic stopping distance of the truck and the tractor trailer." (71, p. 7). This situation is aggravated in wet driving conditions, as "...very lightly loaded truck tires are vulnerable to a pronounced traction deficiency on pavements in which the water cover stands sufficiently above the textural asperities." (69, p. 36).

Finally, abuse of the allowable gross vehicle weight through excessive cargo loadings can exacerbate all of the above problems. A suspension system designed to operate at a certain maximum load will prove to be inadequate at other, higher load levels: "Inadequacy arose through incompatibility of load and vehicle or retaining system, ignorance of dynamic effects and deliberate overloading." (134, p.25). Such overloading can induce unsafe vehicle reactions to road inputs and lead to the failure of suspension components, which can then cause accidents.

4.2 Accidents and Suspension Dynamics

Many commercial vehicle accidents deemed to have been caused by one or a number of the above four factors may also have been contributed to by elements from other sources. Consider the situation where the predominant factor has been identified as driver error: "In many cases the driver doesn't have a choice. No matter how conservatively he drives the vehicle, he may be faced with situations he can't do anything about it." (sic) (76, p.22). Some of these contributory sources may include undesirable dynamic responses on the part of the vehicle's suspension.

When the vehicle's stability and dynamic response characteristics are inadequate or inappropriate, the operator is often called upon to correct the vehicle's behaviour, or at least to attempt to do so. Although an experienced driver may, in some circumstances, be able to minimize the effects of an unstable vehicle, the limited number of available control responses (i.e. steering and braking actions) and the speed at which these responses must be performed do not permit the operator to be very effective in this task. Further, the notion that the proper control of the vehicle is the total responsibility of the operator is questionable: "...if the performance of the vehicle is limited, should the driver be responsible for compensating for that limited performance?" (76, p.22).

In general terms, a vehicle described as being stable or safe is one in which the operator can retain control under normal or predictable driving conditions. Good dynamic behaviour, as described by W. A. Leasure of the NHTSA, is "when the vehicle follows a desired path in response to steering, remains upright or in other words, does not roll over, maintains a limited swept path, and does not oscillate from side to side in an uncontrollable manner." (76, p. 77). From the description of accident factors above, it is obvious that present day vehicles do not always display 'good dynamic behaviour'. Part of the burden of blame for anything less than this 'good dynamic behaviour' must be borne by the suspension system. Therefore, a link can be established between the configuration or design specifications of a commercial vehicle's suspension system and its susceptibility to being involved in certain types of accidents. The types of suspension-related vehicle responses which are considered to be the most susceptible to accidents are the following: "...rollover; trailing fidelity, which includes both the low- and high-speed off-tracking; the rearward amplification phenomena; steering control issues below the rollover threshold; and finally, the oscillatory behavior of multi-articulated trucks." (76, p.79).

Many studies have indicated that slackness or excessive lash in a steel-leaf spring type of suspension can contribute to

easily induced displacements of a vehicle's center-of-gravity when subjected to lateral acceleration forces (100). The result of such displacements is a greater propensity for the vehicle to rollover. Axle springs of excessively low rates can lead to insufficient roll resistance, thus also lowering the vehicle's rollover threshold (91). In contrast, a suspension displaying too great a spring rate can induce torsional deformations in the frame it is supporting. Such deformations can lead to rollovers and/or to poorly controllable vehicle reactions which can also lead to accidents (91). Spring stiffness variances from one axle to another in a commercial vehicle, or, an incompatibility in tractor-to-trailer springing stiffness' can also result in lower rollover thresholds (76).

The topic of trailing fidelity can be considered to include low- and high-speed off-tracking as well as rearward amplification. Leasure defines these phenomena very well: "When we refer to off-tracking, we are talking about the lateral dimension to which the trailer axles fail to precisely track the preceding axles during steady turning maneuvers. Rearward amplification refers to the failure of the trailers to follow precisely the towing vehicles dynamic path during turning maneuvers, such as rapid lane changes." (76, p.87).

Low- and high-speed offtracking originate from two different mechanisms and, as such, have distinctive outcomes or results. The extent to which suspension design could have an effect on the low speed type depends on whether or not axles located at a position other than the traditional steering one permit steering of their wheels. Suspension characteristics such as damping ratios or spring rates would have little or no effect in this case. High-speed offtracking is affected by tire slip angle properties, which are themselves dependent in part upon load distribution as well as the wheel travel direction. A suspension system designed in such a way as to equalize axle loadings among two or more axles could subject the vehicle to varying levels of high-speed offtracking depending on the level of effectiveness of the load equalization. Since high-speed offtracking can lead to accidents through rollovers or collisions with other vehicles (or other objects), a suspension design which minimizes it is preferable to one which does not.

Rearward amplification is usually considered to be of greater significance for articulated vehicles which have two or more trailers. The type of coupling used at the articulation points influences to a great extent the maximum level of amplification which can be developed (76, p.101). However, the loading levels and thus spring rates (or suspension type) used in the coupled trailers also have an effect on the actual level of amplification which is

eventually created (3). A trailer suspension with inadequate or inappropriate damping and/or spring rates could thus, in some situations, determine whether or not a case of rearward amplification caused, or contributed to, an accident.

Steering control or the consequences of such are affected by suspension characteristics in the sense that the latter are often non-linear in nature, and may contribute to the non-linear response of the vehicle to steering inputs. Accidents are more likely to occur in situations where the effect of control function actions are not constant under a large range of operating conditions: "The driver is faced with the seemingly difficult task of controlling a mechanical system which, at low speed, is characterized by long response times, and at higher speeds, is characterized by large levels of turning response per unit displacement of the steering wheel. ... This suggests that the steering controllability characteristics in the heavy truck population may be lacking, and may be a factor in the accident causation process." (76, p.97). Amplification plays a role in this situation, at higher speeds.

Lateral motions or displacements of articulated vehicles are not limited to those engendered through off-tracking or brought about by steering actions: "... articulated vehicles often exhibit oscillatory behavior when simply travelling in a straight line." (76, p.97). The origins of the excitation leading to such oscillations can be traced to a number of sources, including aerodynamic effects or pavement irregularities. Proper suspension design, including spring and damping rates, could minimize the magnitude of these oscillations in order to ensure that they would not lead to collisions with other vehicles: "Although the oscillation may not be sufficiently large to threaten the driver's ability to control the trajectory of his power unit, it may be of concern insofar as the trailing units may encroach on the other travel lanes or intimidate other drivers and thereby cause anomalous maneuvering actions." (76, p.97).

Through the modification of suspension characteristics, it would thus be possible to affect the level of safety, or the propensity towards accident-exposure behaviour, of commercial vehicles. Other measures, such as innovative coupling methods, could also be combined with modified suspension characteristics in order to further increase vehicle stability and safety.

5. HEAVY VEHICLE OPERATING COSTS

Numerous costs are involved in the operation of heavy vehicles, including the infrastructural expenditures involved in the construction and maintenance of roads. Various factors which can affect these costs are outlined in the following sections.

5.1 Infrastructural Costs

According to the Australian Road Research Board's (ARRB) analysis of road infrastructures, the following are the main costs which should be considered or are involved in the construction and maintenance of road systems (128):

1. Actual construction expenditures, including financing charges from such sources as interest or revenue raising.

2. Rent on land not owned by the road authority.

3. Operating costs for traffic control systems.

4. Maintenance costs, according to the minimum acceptable or desired level of deterioration.

5. Rehabilitation or renovation costs not included in the maintenance activities.

6. Reconstruction costs related to enhanced service levels of the road.

7. Residual value of the road at the end of its service life, which is usually considered to be a negative cost.

There are a number of other costs associated with road networks, such as environmental damage and others, but these are usually quite difficult to quantify in monetary terms and will not be considered here.

According to the adopted objective or formula, the methods by which these costs are recovered vary. The ARRB lists a number of these (128):

A. Economic Efficiency.

The classical definition of efficiency is that of a ratio between what is gained and the effort expended for that gain. When examined in an economic sense, the principle of efficiency can be related to the marginal costs of providing goods or services. While marginal costs do not take into consideration the initial investments required for the output of those goods or services, they do indicate the present cost of maintaining the supply of these. In the

public sector, this approach is often used, in conjunction with a larger notion of cost-benefit ratios for the economic community as a whole, in order to determine where and when new or upgraded road facilities should be provided. The marginal cost of the facility is translated into a maintenance cost, which is provided through the implementation of usage charges, while the capital cost is borne by the community (this can be justified as an investment in its future economic development).

B. Cost Recovery.

The principle of cost recovery implies exactly what it says: the recovery of all costs associated with the installation of an infrastructure. An estimate of the total number of users leads to the formulation of a usage charge scheme which attempts to average out all of the costs associated with a facility such as a road. Extra charges are sometimes added in order to raise funds for additional infrastructures or as a means of increasing general revenues.

C. Equity Distribution.

In some cases it may be desirable to subsidize the usage charges of certain users, as a function of larger community benefits, while maintaining the overall level of cost recovery. Therefore, some classes of users, whose value to the community or ability to pay are different from those of the average, may provide or be provided with a subsidy. For example, according to the jurisdiction involved, commercial users may be charged more or less than other road users according to an assessment of their contribution to economic development, traffic congestion, road wear and other considerations.

D. Annual Physical Expenditures.

Considered to be an "administratively simple approach" (128, p. 577), this formula limits infrastructural expenditures to the level of monies actually available in the given fiscal year, while recovery is also limited to those maintenance costs which have been incurred in that year.

The above four formulas are rarely implemented entirely intact, as a mixing of the objectives which each proposes is required in order to satisfy actual conditions. Within the mix which is eventually adopted, the identified infrastructural costs must be assigned to the various classes of road users, usually as a function of the costs which they are assumed to cause or which would otherwise not occur: "The damage caused by trucks to the pavement structure would be a good example of an avoidable cost - it could be avoided if the trucks stopped operating." (128, p.578).

5.2 General Vehicle Operating Costs

Those costs which directly concern the vehicle operator are related to actual vehicle use. The two major categories are:

1. Time dependent, which include driver's salaries, vehicle depreciation rate, taxes, insurance, some types of licenses, and others. Some of these costs may be thought of as fixed for a given period of time, during which representative values for overtime wages and other non-fixed costs can be estimated or assigned to the second category.

2. Use dependent, which include fuel, lubricant, tire wear, general maintenance, major repairs, accidents, certain licenses, toll and road use charges, as well as others.

The ARRB has found that, in non-urban areas, "...the most important variables affecting vehicle operating costs are the geometrical features of the road, the pavement type, the surface roughness and the vehicle operating speed." (128, p.580). Part of this conclusion is backed up by other studies from which they quote: "Paterson (1985) suggests that vehicle operating costs rise by 10 per cent as a paved road goes from good to poor and by 50 per cent as an unpaved road similarly changes. He noted that these costs are usually much more than the associated maintenance costs.." (128, p.580).

Fuel consumption is a major direct use cost factor, which is influenced by the total mass of the vehicle. The vehicle's mass is usually directly proportional to its overall rolling-resistance, its potential for aerodynamic drag as well as to other fuel consumption factors. Several mathematical models have been developed for the calculation or simulation of fuel consumption levels according to the approximate mass of a given vehicle (deterministic, instantaneous, traffic, lumped, urban, rural, and numerous other models). From the fuel consumption values obtained in these models, an approximation of the overall direct use operating costs can be calculated through the use of scaling factors experimentally determined from empirical data.

An example of the distribution of the various types of costs has been calculated by the ARRB as follows (for a six-axle articulated owner-driver truck, reference 128, p.594):

Fixed costs:	51%	from	lease payments	29%
			living allowance	12%
			insurance	6%
			other	4%
Variable costs:	49%	from	fuel	31%
			maintenance, repairs	9%
			tires	9%

As can be seen from the above data, half of the operating costs are of the variable type, and thus subject to reduction. Certain techniques are available for this purpose, such as high-efficiency engines, low wear tires, aerodynamic accessories, and others.

5.3 Active Suspensions and Operating Costs

The previous section has shown that variable operating costs (i.e. direct use costs) constitute a large percentage of the total cost of operating a commercial vehicle. The means of reducing these costs are not limited to the implementation or further development of commonly used techniques. A heavy vehicle's suspension system, for example, is one area where a substantial amount of energy is absorbed. The introduction of a suspension capable of altering its characteristics as a function of road inputs and loading conditions might be able to reduce a heavy vehicle's overall energy requirements, thus reducing its operating costs. A suspension of this type, an "active" one, would ideally be able to vary its spring and damping rates instantaneously, as well as generate displacement forces regardless of the suspension's direction of movement (see Illustrations 3 and 4, p. 48-49).

In order to investigate this subject, an evaluation of the effect of road types on operating costs is first required. An examination of typical commercial vehicle operating costs was performed by Gelpot in 1982. Some of the results of this study were found to be as follows (128, p.594):

COST	PAVED ROAD	UNPAVED ROAD
fuel	35	26
parts, oil and labour	25	28
tires	11	11
capital	15	17
salary	13	18
TOTAL	100 (SIC)	100

These results indicate that fuel consumption assumes a smaller proportion of the overall operating costs on rough roads, while maintenance, capital and salary costs increase. Since the actual amount of fuel consumed on unpaved roads is greater than that on paved roads, it is obvious that rough road conditions impose a significant reduction in operating economy.

The higher fuel costs can be traced in large part to a higher level of rolling resistance due to the poor road conditions. Higher maintenance costs are also induced by such conditions due to the fatigue and eventual failure of equipment subjected to higher road reaction loadings, such as suspension components or vehicle frames. The major objective of an active suspension over such road surfaces

would be to minimize these negative effects. This would be achieved by decreasing the effective rolling resistance and simultaneouly reducing the peak loads and extreme stress reversals associated with rough rolling surfaces.

The installation and operation of an active suspension would incur capital costs and maintenance requirements different from those associated with classical suspension systems. In the case of a hydraulically based system, for example, a hydraulic pump, accumulators and actuating cylinders would replace the air compressor, air bags and shock absorbers or steel springs associated with an air or steel type of suspension.

Although various configurations of active suspensions would be required in order to allow their use on the full range of existing heavy vehicles, the basic components of each system would be common to all. Costs associated with inventory or loss of productive time due to back-order delays would stand to be reduced by such commonalities.

At present, some of the components required in a hydraulic active suspension are considerably more expensive than their counterparts in existing suspensions. Extremely high performance servo-proportional valves, for example, are now available, but only at a cost of approximately US $ 1,200.00 in unit quantities (February 1987 quote). However, as the demand for these valves increases, higher production volumes will result in much lower prices. This has been the case for hydraulic pumps and actuators, the costs of which are now competitive with similar parts in an air-type suspension.

Currently, the extra cost associated with an air-type suspension, as compared to steel leafs, is of approximately $4,000.00 CAN. Under present supply conditions, the extra cost of an active suspension as compared to an air-type suspension would not be much greater than approximately this same amount. Tony Rudd, of Lotus Cars, addresses the cost of active suspensions in the following terms: " The last big barrier to overcome with active suspension is cost. We have got adequate systems for passenger cars but in American terms the package, at the moment, looks like adding $10,000 to the price of a car. But we can see that active suspension is eventually going to cost no more than air conditioning or ABS braking." (136).

The physical configuration of the control system would not be more complex than that of existing electronic equipment presently found on road vehicles, such as fuel-injection hardware (37) or anti-lock brake systems (77,78). As these systems' architecture and chip-count would be similar to those of commonly available personal computers, a cost of the order of CAN $ 2,000.00 or less would be a reasonable estimate for a unit from a non-prototype production run.

Control algorithms of several different types would have to be made available in order to satisfy the dynamic control requirements of commercial vehicles of various configurations. Purpose-written algorithms targeted for use by a particular vehicle would be required in those instances where one had not yet been prepared for a truck of similar characteristics. Eventually, a general-purpose routine with variable vehicle factors could probably be used to prepare a control algorithm, especially in those cases where only minor variations were called for. This method has previously been used to generalize vehicle simulation software (49).

As in the case of the capital costs, maintenance procedures and costs would be different from existing ones. Present-day hydraulic hardware is no more failure prone than corresponding air-suspension type components. Contamination of the actuating cylinder rods or of the servo-proportional valves, as well as damage to the electronic equipment induced by vibrations or shock loads would be the expected main sources of maintenance concerns. Occurrences of the first problem could be minimized through the use and proper maintenance of hydraulic fluid filters, as well as dust bellows and contaminant wiping seals on the actuating cylinders, per accepted hydraulics industry practices. The susceptibility of the electronic system to mechanical damage could be minimized or eliminated by designing the packaging equipment according to the S.A.E. standard J 1455, and performing routine inspections of the vibration absorbing supports and substrates specified therein.

Since a heavy vehicle's active suspension would be more effective at reducing impact loads (124) than a classical one, repair costs related to the vehicle's frame, body, suspension components and cargo could all be reduced, resulting in lower overall operating costs. Heavy vehicles required to use unpaved roads or paved roads in a poor state of repair would stand to gain the greatest reduction in direct operating costs. Because of this enhanced ability of dealing with road vibrations, a sprung driver's seat might not be required, thus offering an opportunity for the reduction of capital costs.

The cost benefits of such a suspension would not be limited to the domain of the vehicle; they would extend into the general economy. A study on the effect of less damaging suspension systems has revealed that "...it is possible to produce heavier vehicles having heavier axle loads, and to operate them without an increase of the road fatigue and without shortening the road life. As a consequence, the goods can be transported more economically with a lower number of trips, a lower fuel consumption and lower environmental loads." (127, p.12). The following section will examine this topic in more depth.

6. ROAD WEAR AND DAMAGE

The wear and damage of roadways is a consequence of their use as a surface for the passage of motorized vehicles. Ideally, the rate at which this wear or damage occurs can be minimized, but the deterioration and eventual need for maintenance or renewal is inevitable. The following will examine this situation and propose some means for an enhanced protection of roads in general.

6.1 Road Damage Types

A recent study has identified five main mechanisms by which the failure or wear of roads can occur: "(1) the failure in fatigue of the substrata; (2) the sub-surface shearing at one or more of the interfaces of the different layers which make up a major trunk road; (3) the cumulative compacting of the running surface and possibly also of the bitumen macadem roadbase; this can manifest itself in 'grooving' or 'tramlining' and reduced surface water drainage; (4) the visco-elastic 'flowing' of the top dressing or even the first of the two layers macadem roadbase material, which is increased by frequency of heavy vehicles passing and road surface temperature; (5) the abrasive wearing of the running surface, which can vary widely since it depends so much on the road stones used." (44, p.51).

There exist several types of easily identified road wear or damage symptoms, each of which is caused by a distinct deterioration mechanism or a number of these in combination, as described above. The following are a few of these.

Pot-holes are a sign of the damage caused by a combination of climatic conditions and road loads. The annual freeze-thaw cycle plays an important role in the formation of this type of pavement deterioration. Ice, expanding as it freezes from water, displaces substrate materials from under the paved surface or expands existing cracks through the various layers which make up the road. Upon thawing, the ice no longer supports the overlying surface, which undergoes flexural forces due to the downward bearing loads of the passing vehicles. Eventually, the depressions so caused turn into holes when the paving material undergoes tensile failure and is dispersed by dynamic wheel forces. Such holes are not necessarily oriented perpendicular to or in parallel to the travel of the vehicles, and can occur across the breadth of the vehicle's track width or along the path of either the left-hand side or right-hand side wheels. These holes are discontinuous and are not necessarily related to adjacent ones, although common underlying substrates and loading effects from vehicles which have encountered preceding holes may induce or encourage the formation of holes in the near vicinity.

Ruts are of a more regular or continuous nature than are pot-holes. The two linear depressions following the average track width of the vehicles which pass above the roadway are directly induced by the deformation stresses imposed by those road wheels. Several studies have analyzed the phenomenon of rutting, and one in particular has formulated a modelling technique for the prediction of its severity (130). Many factors influence the formations of ruts, including the plastic and elastic characteristics of the various pavement, base and sub-base layers, the amount of adhesion between these layers, the respective thickness of these layers, the fatigue life of the asphalt involved, the moisture content of the subgrade, the initial and subsequent in-service compaction levels, the number and average mass of the vehicles using the roadway, as well as a number of others.

Abrasion of the surface is more closely related to the removal or erosion of paving material in particles rather than its deformation or settlement. Some sources of abrasion are tire-to-road slippage, snow-clearing plows, and climatic effects.

Shearing of the paving surface occurs as a deformation or a material loss (of a 'sheet' or thin planar shape) parallel to the surface of the road. Braking and acceleration reactions among other factors are stated to contribute to this phenomenon (48).

Cracking parallel to the direction of travel has been stated to be related to the same factors as rutting (130) while perpendicular cracking may be more closely related to substrate settling or freeze-thaw effects.

6.2 Road Damage Caused by Suspension Dynamics

The following are a set of relative road damage cost figures considered to be avoidable, according to vehicle type, as estimated by the ARRB (128, p.578). In order to facilitate comparisons between them, the first has been used as a base and set to a value of unity.

VEHICLE TYPE	AVOIDABLE COST FIGURES	
	COST/TRUCK-KM	COST/TONN-KM
2 axle rigid	1.00	1.00
3 axle rigid	1.37	0.96
4 axle rigid	1.89	1.09
3 axle articulated	2.11	1.30
4 axle articulated	2.74	1.30
5 axle articulated	3.16	1.30
6 axle articulated	3.95	1.48

From these values, it seems obvious that the avoidable cost related to the damage caused by heavy vehicles to road surfaces is considered to increase as the mass and size of the vehicle increases. In some cases, this cost is believed to constitute the largest portion of road construction expenditures: "Studies indicate that the avoidable costs due to trucks are between 30 and 60 per cent of total costs on arterial routes." (128, p.578). Thus, the collective contribution made by heavy vehicles towards the formation of pot-holes and ruts, as well as to the abrasion, shearing and cracking of the road surface far exceeds that of the population of smaller vehicles.

In an attempt to quantify the relationship between vehicle mass and road damage, the AASHO (American Association of State Highway Officials) performed road test studies which indicated that the level of road damage caused by any given vehicle was proportional to the fourth power of its axle weight. The validity of the exponential factor's magnitude has been confirmed by several subsequent studies. However, recent research has disputed the use of a static axle weight as the base value in the relation: "...damage to the road structure must be a function of dynamic axle loading. This dynamic loading is a function not merely of static load but also of the suspension characteristics of the vehicle, particularly unsprung mass, tyre rate and friction." (48). According to this research, the level of road damage caused by road vehicles would be proportional to the fourth power of the dynamic axle load.

In England, a great deal of work has recently been performed in the study of the effect of higher axle loads on road surfaces. Political pressure from the other member nations of the EEC, most of whom allow 40 tonne GVW trucks onto their roads, has been the main reason for the undertaking of the program. The study, released in 1987 by the TRRL, lists four major factors which combine to cause road damage: "static axle load, dynamic increment, tyre contact pressure and vehicle speed." (44, p.51). While several other contributing factors are mentioned as well, the above four are considered to be the most important ones.[1]

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1 These other factors include: "...damage due to the movement of the road and its substrata, which in turn may be due to changes in retained moisture of the subsoil onto which the substrata are laid, thermal expansion and contraction, frost and de-icing salts, spillage of and penetration by oils, fuel, chemical substances, not forgetting agro-chemicals, and corrosion of reinforcements, non homogeneity of road material, local collapse, edge and buttress breakaway, also settling of pile supports." (44, p.51)

In terms of pavement damage or loading effects from the dynamic reactions of heavy vehicles, the main findings of this study were as follows: "(1) dynamic loading of the pavement is due to the undamped or only lightly damped oscillations of the loaded trailer, as it bounces on its tyres, excited by such small variations in longitudinal profiles as are likely to be found on major trunk roads in the UK; (2) for speeds of over about 30 km/h dynamic loads of up to +15 per cent above the nominal static axle load could occur on tandem axles and that might be +25 per cent on single axle trailers; (3) these dynamic loadings produced stresses and strains in the roadway 20 per cent above those to be expected from the nominal static loads and that consequently the service life of some roads could be reduced by up to 50 per cent." (44, p.51). These figures agree with the AASHO's method of estimating road wear or damage where the fourth power law is used: a reduction in dynamic loading of less than 16% would thus yield a corresponding reduction of 50% in the damage or wear imposed upon the roadway.

A German study performed by the Universities of Munich and Hannover, cited in the TRRL report above, had slightly different results. According to its findings, the greatest damage to the roadway occurred at lower speeds, through a visco-elastic mechanism which allowed the truck tires to 'sink' into the pavement. As vehicle speed increased, the exposure time per unit pavement was reduced, and less damage was apparent. The lack of variation in longitudinal profile (bumps) in the German pavement, as compared to the British one, was probably to blame for this discrepancy (44). This is confirmed by a conclusion of the FAT, whose 1984 report on those German university studies stated: "If there are no dynamic wheel loads, the strain (equivalent to road fatigue) of a horizontal flexible road and of a concrete road as well decreases with increasing speed, up to 40 km/h, then it remains practically constant." (127, p.11). Few dynamic wheel loads will be produced on a bump free surface, which implies that the German roads tested in the FAT study had a 'flatter' profile. North American roads probably have a profile which is similar to or worse than the British one, rather than being as 'flat' or uniform as the German one.

An obvious means of reducing road damage is thus to minimize road loadings. A reduction in general vehicle mass would reduce the static axle load and thus decrease road loadings. A loss of mass significant enough to have an appreciable effect on the level of road damage would nonetheless be difficult to achieve: the cargo weight remains the same, the fatigue life of the vehicle's load-bearing structure must be retained and few high-strength/high-cost materials can be introduced due to economic considerations. Alternatively, increasing the number of axles can reduce the effective unit weight supported by each one. Such a modification, however, can lead to higher levels of friction, which may accentuate

abrasion and shear problems, or to excessive dynamic reactions due to the effect of linked axles. The means with the most promise would seem to lie in the direction of reducing dynamic road impacts through better suspension design.

An important conclusion regarding the relationship between the magnitude of dynamic loads and vehicle or suspension design was stated in the FAT report: " Dynamic load peaks can be relevantly reduced by correct selection of axle and body suspension and dampening." (127, p.12). During the course of an experiment which was performed to investigate the effect of suspension design changes, the FAT researchers found the following: "In the particular case of the selected road, the use of a 11 ton bus axle with adequate suspension adjustment and a harder dampening (which is the essential parameter) did practically not show higher load peaks than the conventional 10 ton axle of a bus from the series production." (127, p.12).

This suggests that the improper design or selection of suspension components can increase road wear. Poorly designed linked axles are an example. Most Canadian provinces require the use of load-equalizing devices between the axles of semi-trailers. These were found to be less efficient than originally believed in the TRRL study: "...due to the inadequacy of the mechanism on the trailer used, which is supposed to share the load between the two axles of the semi-trailer but did not, there could be an increase of 19 per cent in the rear trailer axle load when the road was smooth but had long wavelength undulations." (44, p.51).

Several other studies have shown that suspension design can have a significant effect on the level of road damage. Some of these have demonstrated that different types of suspensions can have greater or lesser dynamic loading effects. A comparison of bus and truck suspensions found the following : "The results of the combined experimental and theoretical study indicate that the softer ($w_n \approx 1.5$ Hz), heavily damped ($\epsilon \approx 0.2$) bus suspension significantly reduces the peak dynamic pavement loads when compared to the stiffer ($w_n \approx 2.5$ Hz), more lightly damped truck leaf spring suspension." (57, p.340).

Another study compared various truck suspensions and found that some were significantly worse than others in terms of dynamic loading: "1. Under smooth road conditions (RCR > 8), all five types of suspensions tested exhibited relatively low levels of dynamic wheel loads, generally in the range from 6% to 12% of static load. 2. Under rougher road conditions (RCR = 5), the dynamic loading coefficient (DLC) increased much more drastically for some types of suspensions than others. At 80 kph, the air bag suspensions

exhibited dynamic loading coefficients of 16%, the spring suspension DLC rose to 24%, while at the extreme, the rubber spring walking beam DLC increased to 39%." (49, p.36).

A reasonable conclusion to infer from the results of the above research programs is that a properly designed suspension system would minimize the level of road damage imposed by a given heavy vehicle.

6.3 Active Suspensions and Road Damage

Many recommendations have been made concerning vehicle and suspension configurations in order to reduce road damage. Some of these involve suspension characteristics which are subject to tradeoffs in relation to overall vehicle safety, handling characteristics, ride quality or other areas.

For example, certain ARRB reports suggest the use of single wide tires as a means of reducing the rollover threshold of certain trucks (81). The FAT report, however, discourages the use of single tires. It states that the contact patch area of the single tire axle configuration is too small to transfer its share of the vehicle load into the pavement at an acceptable pressure level (127).

In terms of road damage another compromise exists. All of the research outlined in the previous section recommends that peak dynamic loads should be reduced in order to minimize road damage. A study concerned with ride comfort has found the following: "The ride comfort of the articulated vehicles can be greatly improved by using low spring stiffnesses. However, lowering the spring stiffnesses below certain values will increase the probability of suspension bottoming. The choice of suspension spring stiffnesses, therefore, is a compromise between riding comfort and road safety." (98, p.55).

These compromises arise out of the requirement of selecting suspension characteristics which, under normal operating conditions, will not be adjustable in order to meet varying loading or road conditions. The selection and specification of damping and spring rates is thus achieved by attempting to obtain, for a given road speed and road profile, an acceptable mix of such factors as vehicle stability, riding comfort, load capacity and minimized operating expenses. In most cases, the reduction of road damage is not a main consideration of fixed-rate suspension systems.

An active suspension, however, could eliminate the need for compromises of this nature, while at the same time decreasing road damage by reducing peak dynamic loads. In contrast to passive systems, an active one would be able to

adjust its effective spring and damping rates according to varying loading and road conditions.

In a classical suspension, the wheel reactions are a function of the fixed spring and damping rates, as well as of the road excitations. Subjected to the same road surface, an active system would make its road wheels follow road excitations much more faithfully than a passive one. It would do so by setting its effective spring and damping rates according to the nature of the excitation and, if required to, by exerting a displacement force on the wheel. The objective would be two-fold: to manage and absorb the road surface energy presented to it, as well as maintain the vertical position or travel height of the suspended mass in order to avoid disturbing the vertical inertia of the vehicle. Performed well enough, this latter function would pre-empt the formation of any significant reaction forces, and would thus reduce road wear as compared to a classical suspension system.

There exists proof to demonstrate that such a reduction can be realized. Experimental studies of airframe landing loads have shown distinct advantages for an active system: "Aircraft undercarriages double as a suspension system and an energy absorption system. N.A.S.A. research (Ref: 2) indicates that a 20-30 per cent reduction in structure loads on landing are possible with active control." (32, p.26). Research performed in England on semi-active landing gear systems for aircraft has yielded similar reductions in structural loadings, as compared to passive systems (28).

An equivalent active suspension system installed on a road-vehicle would be expected to yield similar reductions in structural loadings. Conveniently, any such reductions in axle or frame reaction loads can also be directly related to an equivalent reduction in the dynamic road loads, as stated in the FAT report (127). By means of the AASHO's fourth power law for road wear, a reduction of 20% to 30% in dynamic structural loads would thus yield an estimated 59% to 76% reduction in road wear. Considerable savings in road maintenance expenditures could thus be expected as a direct result of the use of active or semi-active suspensions on heavy vehicles. Such savings could be used for the extension or upgrading of existing transportation infrastructures or other community improvements, including mass-transit systems.

7. HARDWARE REVIEW

Most engineering design projects base themselves, at least in part, on an array of existing equipment. Otherwise, the analysis and fabrication requirements in terms of both labour time and capital investment would be excessive. This is especially true in the case of low-volume or limited production items, as in the introduction of a new type of heavy vehicle suspension system.

The practical means of actively controlling a heavy vehicle's dynamic responses are limited, for a number of reasons. For example, the correct response to a given excitation must be available within a very short time delay, a delay which must effectively decrease as the vehicle's speed increases. If the given response involves the generation of a force, the actuator providing it must be of a non-slip or positive nature, in order to lock-up the suspension or create displacements opposed to very high dynamic loads.

A convenient method by which the first condition may be satisfied is through the use of high-speed, digital micro-electronics: mechanical controls would be too slow, while analog electronic controls would be extremely complex to design or modify.

A number of alternatives are available for the actual generation of the controlled response forces, including systems driven by electrical, hydraulic or pneumatic power. Among these, only a hydraulic system can offer truly positive actuation, as both magnetic fields and gases are subject to compression. Hydraulic suspension systems of various configurations have been used in production passenger vehicles for over three decades. As electro-hydraulic control applications are common in present-day manufacturing industries, there exists an extensive equipment inventory for the design and implementation of a hydraulically powered active suspension system.

7.1 Existing Hydraulic Equipment

The main hydraulic components which would be required for the implementation of an active suspension are: pump, accumulators, actuators, servo-proportional valves, oil coolers, oil filters, oil reservoir, oil lines and line fittings. Existing manufacturers of these include the following:

PUMP:
- -Bosch-Hydraulik
- -Commercial Shearings, Inc.
- -Rexroth Corp.
- -Bendix Fluid Power Division

-Webster Electric Company
-Commercial Shearings, Inc.
-Hydroperfect International
-Roto-King Pump (Houdaille)
-Parker Fluidpower
-Oildyne
-Dynex/Rivett
-Prime Mover Controls Inc. (CANADA)
-Cross Manufacturing, Inc.

ACCUMULATORS:
-Bosch-Hydraulik
-United Technologies Fluid Power Systems
-Christie Hydraulics Ltd.
-Sperry-Vickers
-Incom International Inc.
-Parker Fluidpower

ACTUATORS:
-Oildyne
-Commercial Shearings, Inc.
-Parker Fluidpower
-Midwest Hydraulics
-Miller Fluid Power
-Incom International Inc.
-Fluid-Pack International Limited. (CANADA)
-H.E.S. (CANADA)
-Nopak Inc. (CANADA)
-Brunning Hydraulics (Imperial Clevite Inc.)
-DHB - Componentes Automotivos S.A.
-Cross Manufacturing, Inc.
-Lehigh Fluid Power, Inc.

VALVES:
-Moog Inc.
-Rexroth Corp.
-Fluid Controls, Inc.
-Amca International Corp.
-Brunning Products (Imperial Clevite Inc.)
-Waterman Hydraulics (Imperial Clevite Inc.)
-Eemco (Datron Systems, Inc.)
-Hydraulic Servocontrols Corporation
-Dynamic Valves Inc.
-KIP Incorporated
-Vickers, Incorporated
-Parker Fluidpower

OIL COOLERS:
-Sperry-Vickers

OIL FILTERS:
-UCC Amfac/Rucker Products
-Cross Manufacturing, Inc.
-Hydroperfect International
-Filtroil Corporation
-United Technologies Fluid Power Systems
-Sperry-Vickers
-Parker Fluidpower

RESERVOIR:	-Hydroperfect International -Sperry-Vickers -Incom International Inc.
OIL LINES:	-Eaton Corp. -Imperial Eastman
FITTINGS:	-Brunning Hydraulics (Imperial Clevite Inc.) -Imperial Eastman -United Technologies Components Division -Maeward, Inc.

7.2 Existing Electronic Equipment

The major electronic components and transducers which would make up the sensing and control systems of an active suspension are the following: microcontrollers, cylinder-oil pressure sensors, acceleration sensors, angular tilt sensors, road-wheel speed sensors, road-wheel height sensors, steering wheel position sensors and high-speed data lines. Existing manufacturers of these are:

MICROCONTROLLERS:	-Motorola -Intel -NEC -OKI Semiconductor
PRESSURE SENSOR:	-VDO Instruments, Inc. -Techmar Incorporated -Raychem Limited -Entran Devices, Inc. -Robinson-Halpern Company -Transamerica Delaval -Sensotec -Validyne Engineering Corporation
ACCELERATION SENSOR:	-EntranDevices, Inc. -Colombia Research Labs, Inc.
TILT SENSOR:	-Spectron Glass and Electronics, Inc. -Colombia Research Labs, Inc. -Diesel Kiki Co., Ltd. -Robinson-Halpern Company -Sperry -Signal Systems International Inc.
SPEED SENSOR:	-Diesel Kiki Co., Ltd. -VDO Instruments, Inc. -Dynamics Research Corporation -CTS Corporation -Airpax

HEIGHT SENSOR:	-Diesel Kiki Co., Ltd. -Dynamics Research Corporation -Systron Donner (Thorn EMI) -CTS Corporation -Robinson-Halpern Company -Celesco Transducer Products, Inc. -New England Instrument Company -Spectrol (United Technologies) -Alps Electric (USA), Inc. -Eaton Corporation (Controls Division) -Oriel Corporation -Penny & Giles Inc. -Waters Manufacturing, Inc.
STEERING SENSOR:	-VDO Instruments, Inc. -Dynamics Research Corporation -Farand Controls
DATA LINES:	-Toshiba Corporation -United Technologies Automotive Div.

8. ACTIVE SUSPENSION DESIGN

The electro-hydraulic type of active suspension alluded to in the previous section is more closely examined here. A few alternative means of achieving the effect of an active suspension are presented and evaluated.

8.1 Active Suspension: Mechanical Configuration and its Governing Parameters

As for any suspension system, the purpose of an active one is to absorb and/or manage forces and displacements. These would otherwise be directly transferred into the sprung mass of a road vehicle passing over road surface irregularities. Several compromises must be made in the design of a classical vehicle suspension, some of which were examined in section 6. An active suspension, however, is not subject to such compromises, as its effective springing and damping rates are variable according to changing vehicle and road conditions.

The spring of a classical suspension acts as a storage buffer for road forces. The damping present in the system governs the rate at which these forces are transferred to the vehicle or returned to the road. Spring and damping rates are specified as a function of the usual operating mass and velocity of the vehicle, as well as of the average road surface profile characteristics. Thus, a vehicle equipped with a classical suspension is ideally suited for operation only within those conditions which correspond to the specified ones.

An effective active suspension would set the vehicle's springing and damping rates according to the ride requirements imposed by instantaneous changes in load, velocity or road roughness. Appropriate operating characteristics for a wide range of conditions would thus be available. This implies that the suspension components would have to be able to absorb and exert large forces, through varying displacements, at high frequencies.

A number of alternative technologies are available in order to accomplish this, although most have notable shortcomings. Electrically powered actuating systems can exert forces and create displacements at high frequencies. However, they do not exhibit sufficient positive thrust at a reasonable level of power consumption or mass at the present time. The near-term development of amorphous metals or superconductive materials yielding high levels of magnetic flux in exchange for low levels of power input may alleviate such problems.

Due to the inherent compressibility of their working fluids, pneumatic systems also suffer from positive displacement

limitations. Hydro-pneumatic systems are particularly applicable to semi-active systems, but suffer from the same lack of positive displacement in general service as do the pneumatic ones.

Electro-rheological fluids are incompressible and have the distinct advantage of allowing the use of inexpensive valves having frequency response capacities approaching those of solid-state devices. These types of fluids are extremely promising for use in active suspensions of the near future (15 to 20 years), once they are available in commercial quantities and at a reasonable cost.

Given the present state of technology, the most practical means of satisfying the force and displacement requirements of an active suspension would be through the use of a positive displacement hydraulic system which is managed by a digital, high speed electronic control system.

Previous hydro-pneumatic type suspensions have demonstrated the use of hydraulic actuating cylinders as a means of transforming linear forces and displacements into fluid pressure and flow, or vice-versa. For the purpose at hand, this forms an elegant and appropriate solution. A hydraulic active suspension's actuators would be placed in the same location as the components of a classical suspension, that is, between the vehicle's body and the axle supporting the road wheels. A system capable of equalizing wheel loads or of resisting lateral accelerations would require an independently controlled actuator to be present at each end of the axle.

In order to emulate the behaviour of an ideal spring and damper assembly, an actuator would have to resist or exert forces of various values at various rates. A positive displacement device such as a hydraulic cylinder could do so by adjusting the pressure and flowrate of the hydraulic fluid directed to either side of its working piston. This piston naturally tends to move towards the direction in which there is a lower level of fluid pressure. By exploiting this feature, a close control of fluid flow would also control the amount and velocity of retraction or extension.

When subjected to a bump, the spring in a classical suspension becomes compressed at a rate regulated by the damper. An actuator would simulate such a compression by means of retracting, i.e. by reducing its overall length. A valve would direct additional fluid under pressure into the chamber 'below' the piston while, simultaneously, the fluid in the chamber 'above' the piston would be allowed to escape.

The actual flowrates used to emulate the spring and damping rates would be obtained by regulating the throat openings of the valves leading to the actuator. High flow rates during retraction would simulate low spring rates and low damping, while high flow rates during extension would emulate a high spring rate and low damping (or overdamping under certain circumstances). Variations in valve openings during the course of a retraction or extension would simulate a spring and damper assembly having continuously variable rates (see illustration 5, p. 54). A cessation of fluid flow would render the suspension inactive, as if the wheels or axles were solidly affixed to the vehicle body or frame.

Valves capable of operating at high frequencies, as well as a means of controlling them, would be required in order to create such a closely regulated flow. A continous supply of hydraulic fluid under pressure would have to be supplied to the valves as well. The source of this would probably be an engine-driven hydraulic pump. In order to limit the amount of energy consumed by this pump, its output could be regulated by the same control system managing the valves. Other mechanical accessories associated with the hydraulic circuit would include filters, coolers, and oil lines.

A successful implementation of an active suspension system on a passenger vehicle has already been made by using mechanical and hydraulic components as described above (124). A similar mechanical configuration could be used for the implementation of an active suspension on heavy vehicles. Buses, for example, could use the same basic configuration as a passenger car, i.e. a central hydraulic pump supplying all of a bus' valves and actuators which, in turn, would be managed by a single electronic control system. Articulated vehicles require a different approach, however, due to the interchangeability of tractors and trailers (see illustration 6, page 55).

Air powered brakes for trailers have been universally adopted because of the fact that they do not have to be purged or 'bled' whenever the trailer is disconnected from one tractor and reconnected to another. The same would not be true of hydraulically actuated brakes, or, of other accessories designed for use with non-compressible fluids. A trailer equipped with a hydraulically powered active suspension dependent on a source of pressurized liquid from the tractor would require an additional operation when switching tractors: the purging of its supply circuit. Further, this type of trailer could only be hooked up and operated with a tractor equipped with such a hydraulic system.

In order to retain the articulated vehicle's advantage of being able to interchange prime movers and trailers, each of these should be equipped with a suspension system

independent of specialized interconnecting equipment. Thus, in the context of active suspensions, both the tractor and the trailer would have its own control system and the trailer would be equipped with a non-hydraulic power supply system. An active suspension equipped trailer would thus be able to operate with an ordinary tractor, or, vice-versa.

Refrigerated trailers already have their own source of power for cooling purposes: a small, trailer-mounted diesel engine driving a refrigerant compressor. A hydraulic compressor could be driven by either this type of engine or the compressed air from the trailer's general air circuit. Either of these solutions would allow the trailer to have an operating active suspension regardless of the type of suspension present on the tractor.

The transformation of compressed air into hydraulic power, or, the use of an auxiliary engine to supply that power can have a number of drawbacks, including higher maintenance costs. An alternative which would eliminate this equipment and rely instead on an intermittent compressed air supply in order to non-instantaneously vary spring stiffness and ride-height is a semi-active suspension. If sufficient suspension travel is available, a study has shown that "...semi-active (dissipative) systems employing passive springs to support the body mass and rapidly variable dampers can be almost as good as fully active systems." (29, p.227). The use of a similar mechanically controlled semi-active suspension on a racing car has demonstrated these performance capacities (56).

A semi-active suspension on a trailer would require its own control system and sensor network in order to regulate the valve openings in the dampers as well as vary its spring rates in a non-instantaneous manner. Such a suspension would adjust its spring stiffness as a function of a varying load or of slow dynamic events of the order of 1 Hz. Any tractor with a normal air supply line could thus be used to tow a trailer equipped with this type of suspension.

8.2 Active Suspension: Electrical Configuration and its Governing Parameters

A control system designed to manage the functions of an active suspension's valves, pump and other hardware must be endowed with extremely rapid response capabilities. In addition to this, it must be able to sense what is happening to the vehicle as well as assess the road conditions present under each suspension actuator.

An inertial control system was used in early attempts at implementing an active suspension. A double-decker bus in England as well as a few passenger vehicles were equipped

with such a device (35, 89, 99). The results obtained were not entirely favourable, especially with regards to the rapidity and accuracy of the system's responses. Partly due to the nature of the control system, hydraulic fluid-flow was found to be quite high and thus power consumption of the pump was considered to be excessive.

A high-speed microelectronic control system could have resolved the delay problems encountered by an inertial system. Such a system could easily be linked to sensors informing it of road and vehicle conditions on an instantaneous basis. Following the processing of this data, valve opening settings could be determined and sent to the actuators along high-speed data lines.

Wheel or axle loads would be assessed by means of a pressure transducer located in the actuator or valve body. Sudden gains in pressure would result from the presence of a bump at a wheel, of a lateral acceleration imposed on the vehicle, or, of an inclination encountered while going up or down slopes. Further data sent to the control system by accelerometers and inclinometers would permit it to identify the sources of changes in actuating cylinder pressures. Valve openings at various cylinder locations would be modified according to these values.

Suspension height sensors would allow the control system to measure ride height and maintain it at a prescribed level. Under static conditions, these sensors would permit the sampling of axle loads at the normal ride height. Under dynamic conditions, readings of instantaneous axle positions would allow the calculation and implementation of incremental spring and damping rates. An important purpose of these incremental rates would be to reduce suspension displacement velocities near the limits of suspension travel. This would prevent the suspension components from colliding with their bump stops, a situation which would otherwise cause damage to the suspension as well as introduce higher shock loads into the vehicle's frame.

Wheel speed sensors would allow the control system to identify situations where wheel-lockup or slippage might be influencing actuator pressure or accelerometer readings. A steering wheel sensor, when used by the control system in conjunction with road speed information, could give advance warning of handling manoeuvres likely to result in unstable vehicle behaviour such as rollovers. This would allow the control system to stiffen or lock certain actuators in order to increase the vehicle's instantaneous rollover threshold.

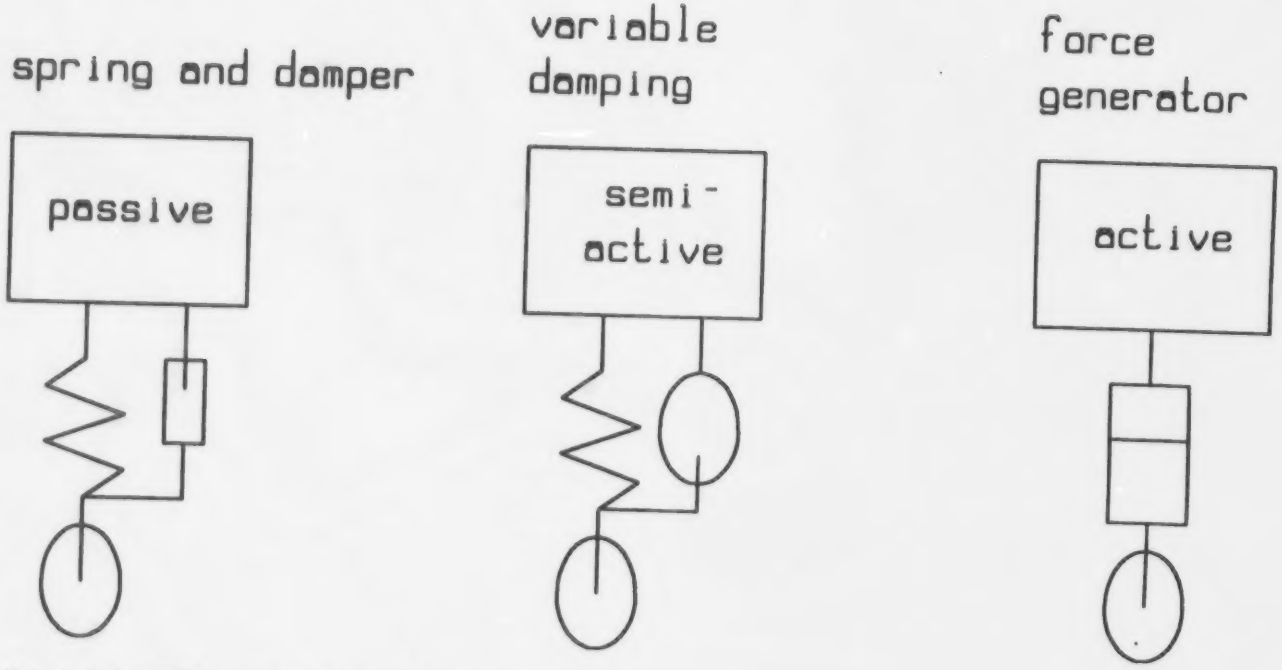

ILLUSTRATION 3: Passive, semi-active and active models

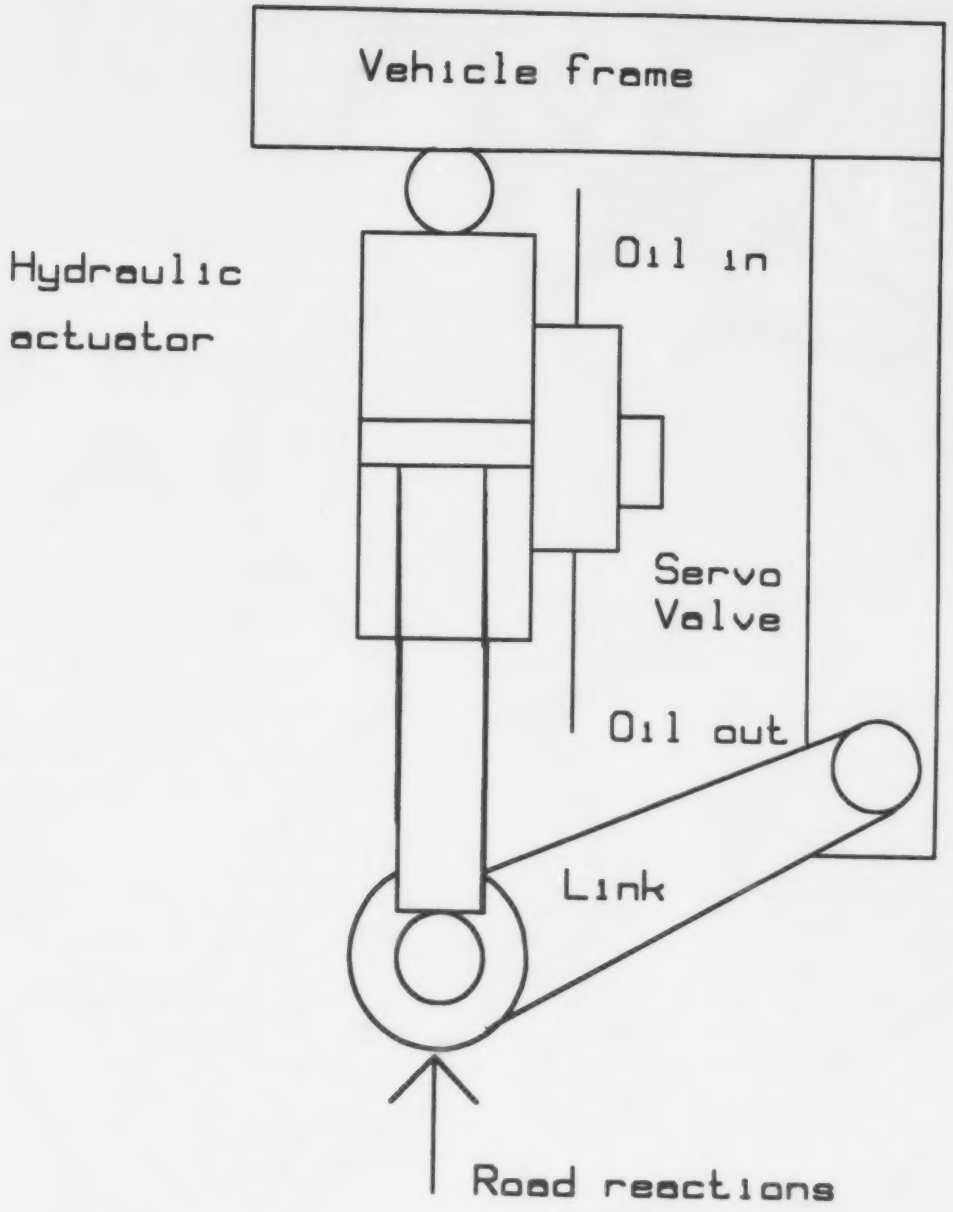

ILLUSTRATION 4: Example of a force generator

8.3 Mechanical Parts Specification

The specification of an actuator intended for a heavy vehicle's active suspension system depends on several criteria. It cannot be selected until the mechanical configuration of the suspension links has been established. These links may make use of bellcranks or other devices which, for example, place the actuator along or across the frame of the vehicle. According to the kinematic characteristics of these links, the loads and displacements experienced by the axle will not necessarily be the same as those transmitted to the actuator. It may thus be called upon to deal with shorter or longer stroke displacements, as well as higher or lower loads, than those directy imposed on the wheels by the road surface.

In some cases, the actuator may be placed directly in line with axle displacements. The actuator's stroke then corresponds directly to the suspension's total travel. Each actuator is expected to support half of the axle load, therefore the piston's surface area will be a function of this half-load value and the working pressure of the hydraulic fluid. The use of accurate numbers for the load and stroke values will help to reduce fluid flowrates and energy consumption by minimizing the volume of fluid to be pumped for a given wheel displacement. Nominal values for suspension travel and hydraulic fluid working pressure would be of the order of 250 to 500 mm. and 200 bar, respectively.

High-performance servo-proportional valves are available at the present time. Satisfactory results have been obtained in passenger cars with valve units having response times of 250 Hz and flowrates of approximately 40 litres-per-minute (124). Although a higher valve response frequency may be desirable, say 500 Hz at a 90 degree phase-lag, the flowrate of 40 l.p.m. should be satisfactory unless a high mechanical ratio has been used to reduce the axle displacement into a very short stroke. Excessively short equivalent strokes lead to a large multiplication of the applied load, which then requires a large piston area. If the piston area is too great, the actuator chamber volumes may then call for valves with a higher flowrate capacity.

Pump flowrates approximately equivalent to those of the valves have proven to be sufficient in passenger vehicle experiments. Since a heavy vehicle's active suspension pump would be called upon to service a larger number of actuators, it would be reasonable to expect a proportionately higher capacity to be required. For the purpose of initial trials, a pumping capacity of approximately 10 l.p.m. per actuator would serve as a guide in the selection of a pump.

The selection of other components such as oil lines, accumulators and oil coolers can be roughly based on the above pressure and flow values. The volume of the oil reservoir will depend on a number of factors, such as available space, but a reasonable nominal value is of the order of 10 litres.

In the case where a semi-active suspension is specified, the performance criteria and specification methods for the actuators, valves, accumulators and the other equipment will remain similar to those stated above. A larger accumulator would be required in order to transform air pressure and volume changes into hydraulic fluid displacements. Other accumulators would be used as variable rate springs adjacent to the actuators.

8.4 Electrical Parts Specification

The control system must be designed to react as quickly as it can to the information delivered to it by its transducers. One means of doing this is by minimizing the number of discrete components which are used to create it. Making use of microcontrollers instead of microprocessors can assist in this, as the former are already equipped with memory and analog-to-digital conversion facilities. The transfer of information between various functional areas of a microcontroller can thus be performed extremely rapidly.

Eight-bit microprocessors were used in the implementation of a control system for a passenger car's prototype active suspension, where four actuators and their associated sensors were serviced by the controller (125). Present buses, tractors and trailers do not require a control system capable of managing more than eight sets of actuators and sensors. The sixteen-bit microcontrollers now available are much faster and have a much greater capacity than the eight-bit microprocessors which were used in the above prototype. This makes them capable of handling the frequency response performance requirements of a heavy vehicle's active suspension: the sampling of 8 sets of 2 to 4 parameters at 500 Hz.

Pressure sensors should have a range of the order of 0 to 250 Bar, with a resolution of approximately .1 Bar.

The accelerometers should be able to detect from 0 to 1 G., with a resolution of about .01 or .02 G.

Inclinometers can be limited to a range of 0 to 60 Degrees, with a resolution of .5 Degrees.

The total range of the axle or wheel height sensors depends on the total suspension travel or the overall actuator

stroke length. An example of a nominal value is 250 mm., with a resolution of 2 mm.

Wheel speed sensors should be able to sense rotation from 0 to 1000 rpm, with a resolution of 1 or 2 rpm.

8.5 Active Suspension: Software Configuration and its Governing Parameters

Although the mechanical and electronic obstacles to the implementation of an active suspension for road vehicles have been eliminated through the development of high frequency response proportional valves and high-speed microelectronics, there remains the question of control instructions and philosophies to be addressed.

In a prototype active suspension system developed for a passenger vehicle, the control software was modelled on a whole-vehicle basis: "It was, however, decided to simulate 'modal' rather than individual 'corner' springs, largely because a previous theoretical investigation had revealed fundamental difficulties in optimising damper settings for a conventional vehicle." (32, p.24). Other theoretical studies have asserted that distinct wheel actuators can satisfactorily emulate ideal suspension springs and dampers. There are thus several ways of approaching the management of an active suspension.

A substantial body of theoretical work concerning the control methods of active suspensions has been written. Some have recommended the use of a deterministic, linear control philosophy with a preview of the road surface profile ahead of the vehicle (2,27,96). A stochastic optimal control is favoured by others (54). The modal spring philosophy is achieved through a decoupled approach (4). Others have chosen a Ricatti state controller for optimization, using input processes with time delays (5). Several other control schemes have been proposed (14).

Which control philosophy is the best one? Which one should be selected for a heavy vehicle active suspension? These questions are not of a trivial nature. The community of dynamicists is not in agreement on this subject, and find it difficult to compare their results: "Even when all practical considerations are neglected temporarily, and modern optimal control theory is used, the results of various investigations are not always comparable. ... Even when similar control concepts are studied, if different design philosophies or, mathematically stated, different criteria are used to judge suspension performance, the results for optimum designs may not be comparable" (58, p.341)

Notwithstanding the controversy over controller type, an initiation routine and general behaviour characteristics are called for in any control system. These are usually contained in a software loop which acts as a kind of operating system. In this loop are a number of evaluations and instructions, as well as the controller algorithms, which determine the actions to be made by the mechanical equipment. Given the mechanical and electrical configurations described previously, the form of a pseudo-code version of such a routine would resemble the following:

```
1.Parked:
          Ignition key out
             -set actuators to locked by closing valves

2.Startup: Ignition key inserted, engine off
             -read suspension height at each wheel
             -read pressure in each actuator
             -read inclinations
             -read pressure left in the supply accumulator
             -calculate static load distribution
             -calculate static height corrections in case of
                        fluid leakage: alert operator.
             -calculate initial pump flow and pressure
           Engine on, no gears engaged
             -set actuator valves if static height
                correction is required
             -set pump output and pressure
             -display load values to operator

3.Gear engaged:
          Check Speed: if speed < .1 km/h
             -set actuators to locked by closing valves
             -reiterate startup routine
          Check Speed: if speed > .1 km/h
             -read suspension height at each wheel
             -read pressure in each actuator
             -read inclinations
             -read all other sensors
             -calculate instantaneous valve settings
             -set valves
             -reiterate until speed conditions change
```

Throughout the operation of this loop, stability and safety checking calculations would be made, in order to adjust the dynamic characteristics of the vehicle on an instantaneous basis. Some of these are outlined in the next section.

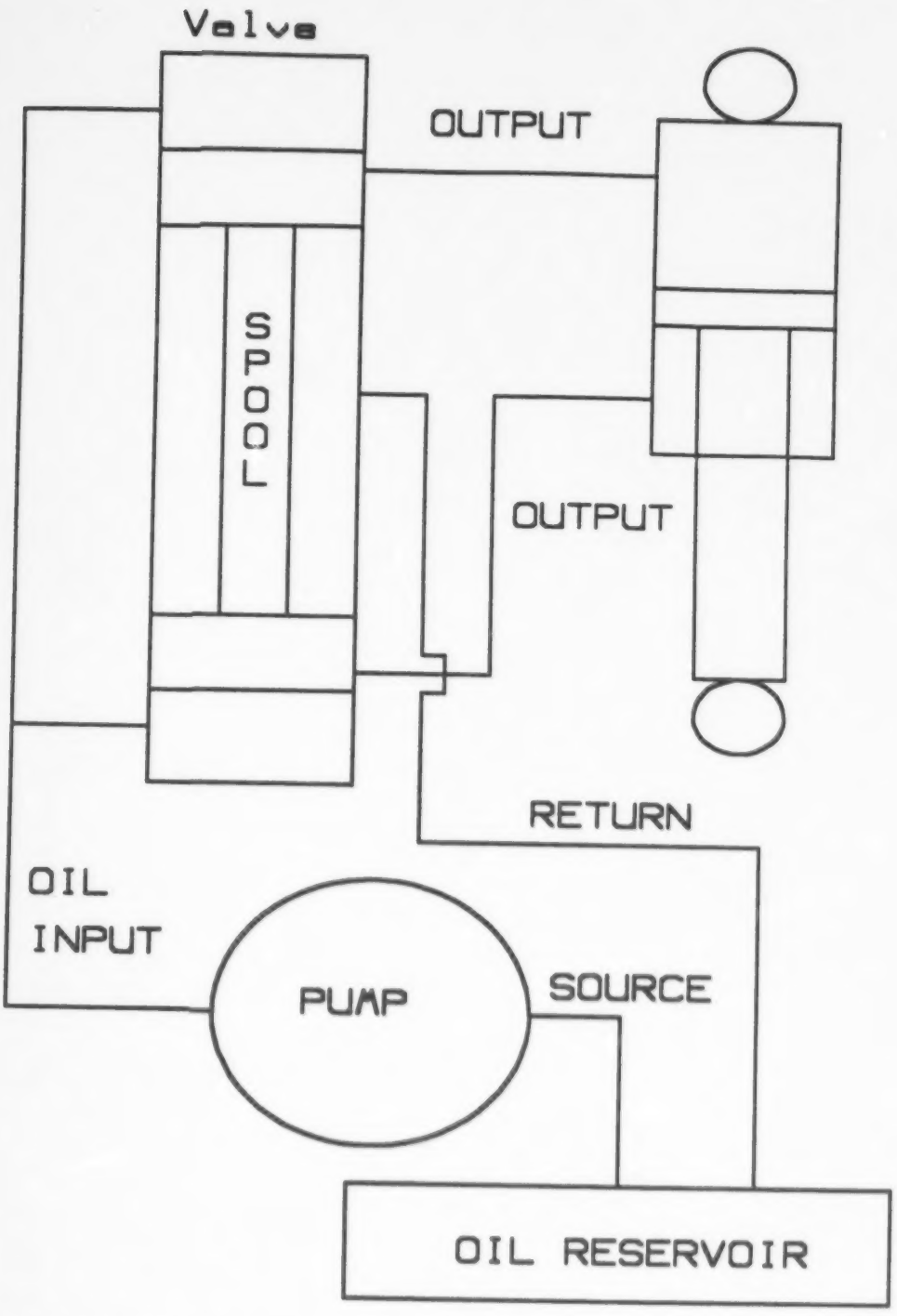

ILLUSTRATION 5: VALVE

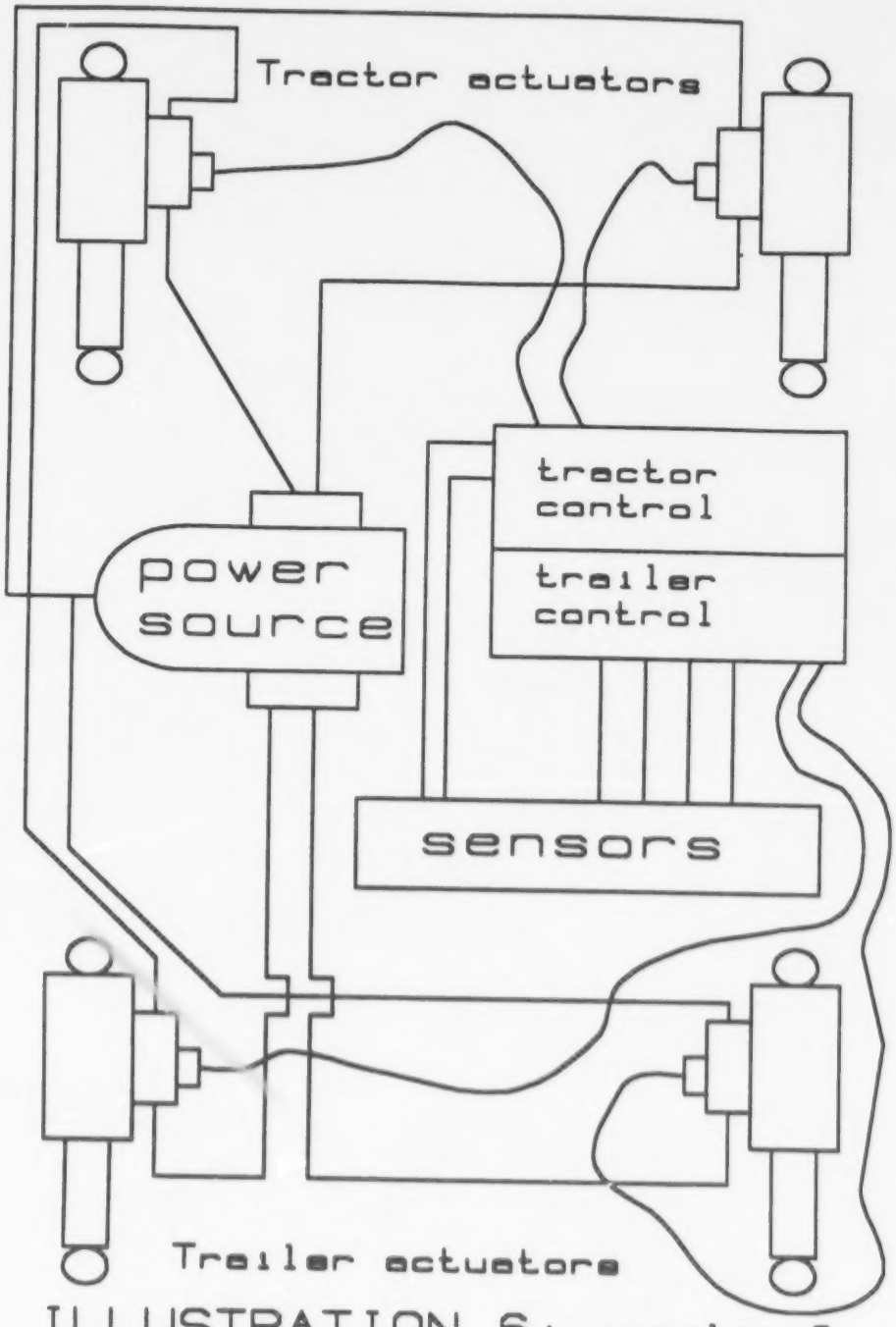

ILLUSTRATION 6: example of an active suspension for an articulated heavy vehicle.

9. ACTIVE SUSPENSION DYNAMICS

The main purpose in adopting an active suspension is to modify the dynamic behaviour or characteristics of a vehicle. Benefits in terms of enhanced safety, reduced road damage and energy savings are sought from such modifications.

9.1 Identification of Safe Dynamic Behaviour, and the Effect of Active Suspensions

As seen in section 4, the suspension characteristics of a heavy vehicle can have a significant effect on its stability and safety characteristics. However, in some cases the modification of a suspension in order to obtain a higher rollover threshold can compromise its road damage characteristics. The introduction of active suspensions can eliminate this compromise as well as others.

An active suspension could be programmed to operate in the safest mode possible for the given load, speed and road conditions to which it would be subjected. It would not be able to compensate for operator inattention, although it could assist in recovering vehicle control in the case of slight driving errors. A warning system could use information directly available from the active suspension's control system in order to alert the driver of dangerous handling situations. This has been suggested in conjunction with the heavy vehicle safety study mandated by the United States's congress: "We may need to look at the aircraft industry. In any airplane there is a stall indicator, either visual, audible or both. There is a turn and bank indicator which tells the pilot whether or not he is doing things properly. ... It would be nice if we had this type of instrumentation on trucks." (76, p.130).

Some of the conditions which are considered to contribute to vehicle rollovers include excessive cornering speeds, inappropriate steering manoeuvres, and the specification of excessively soft suspension springs. A study has shown that replacing the compliant elements (springs) in a truck suspension with solid ones (a locked suspension) can increase the vehicle's rollover threshold by 40% (69). Causing the truck to lean inwards in such circumstances could increase its roll resistance by an even higher margin. An active suspension could thus attempt to prevent the incidence of a rollover either by effectively stiffening its spring rates, by locking itself, or, in extreme cases, by raising its actuators on the outer-side of the vehicle and lowering them on the opposite side, thus inducing a banking effect.

In order to act in this fashion, the suspension's control system would have to be able to recognize a situation as being potentially unstable. It would do this by examining information about current dynamic conditions (lateral accelerations, imposed load changes to the road wheel actuators, steering wheel position and other sensor values) and comparing it to previously recorded data. If excessive changes were found to occur over the time interval involved, or, if a trend towards excessive values was developed over a number of such comparisons, then the system could implement corrective action. The nature of this corrective action would depend on the perceived situation, and could be selected through the use of a look-up table or a sub-routine specifically written for a given situation, such as a roll-over prevention routine, a resonant behaviour prevention routine or others.

Many of the accident situations described in section 4 could be dealt with in this manner, such as ramp-related accidents, trailer hang-ups, load shifts, rearward amplifications and others. Road profile induced accidents could also be prevented, as the reduced impact of an active suspension would reduce the incidence of component failures, and the variable spring and damping rates would prevent resonant behaviour leading to loss of control.

Some safety problems have been traced to incompatibilities between tractor and trailer suspensions. An active suspension, either on the tractor or on the trailer, could adjust itself to the spring characteristics of the other, thus reducing the effect of such incompatibilities (76, p.59).

An active suspension could also assist in enhancing the distribution of braking efforts. As noted elsewhere, linked axles do not always share their loads evenly. Under braking, this situation is exacerbated: "As expected, four-spring suspensions tested are found to transfer load from the front axle onto the rear axle during the application of brake force. Interestingly, the magnitude of interaxle load transfer has also been found to decrease with increasing negative pitch angle." (98, p.,27). An active suspension would prevent an uneven transfer of loads under such conditions, allowing each wheel to develop an optimum amount of braking force.

9.2 Rolling Resistance Sources and the Effect of Active Suspensions

Rolling resistance has often been identified as an important source of energy loss and the cause of increased fuel consumption for road vehicles. In most studies, rolling resistance was considered to be directly related to the

energy lost in deforming the tires (hysteresis losses) and the rotational friction of the wheel bearings. However, travelling over rough road surfaces can require substantially more energy than that expended through simple rolling resistance.

The very presence of damping units in a vehicle suspension implies that there is energy being absorbed through wheel or axle movements. The amount of energy actually being expended in this manner can be substantial. One study reported that: "Rough roads lead to increases in rolling losses as large as 20 percent and, in addition, cause an increase in aerodynamic drag." (24, p.378). Another determined the following: "Computations and measurements indicate that the additional energy losses attributable to the traversal of a typical primary road at a speed of 13.33 m/sec (30 mph) are of the order of 10 per cent of the rolling energy loss occurring on an absolutely smooth surface." (7, p.170).

Several mechanisms have been proposed to explain how a rough road surface can contribute towards total rolling losses. Some compare rough road surfaces to a series of incremental slopes, and that the energy required to scale these are related to rolling losses. Others believe that the lost energy can be traced to the impact forces dissipated in the tires following the reaction of the suspension to road surface irregularities.

An active suspension could show considerably less rolling losses of this nature, as its typical road reaction forces have been shown to be much lower than those of classical suspensions. The fact that suspension characteristics can affect the level of rolling losses has been established in another study, which concluded: "Depending upon the terrain profile and desired speed, it seems clear from this study that a tuning of the suspension could either increase or decrease the propulsion power requirements." (26, p.152)

9.3 Advantages of an Active Suspension

In short form, the following are the main advantages which would be gained through the adoption of an active suspension system for heavy vehicles:

-Decreased road wear and damage (approximately 50%).

-Enhanced safety (potentially 15% less accidents).

-Decreased rolling loads and resistance (approximately 10%).

-Enhanced braking effort distribution.

-Optimal spring and damping rates at all loads.

-Loss of wheel-hopping and resonant axle behaviours.

-Decreased vehicle fatigue and tire wear.

-Decreased operator and passenger fatigue.

-Automatic static levelling.

-Automatic dynamic levelling.

-Operator selected ride level: easier passage through snow or rock-strewn roads at high ride level, better aerodynamics at low ride level, loading dock compatibility.

-Operator information: display of loads, lateral acceleration warnings, self-diagnostics.

10. CONCLUSIONS AND RECOMMENDATIONS

Conclusions:

The technology for a practical, effective and moderately priced active suspension is now available.

The potential benefits of the use of an active suspension on heavy vehicles are important and widespread. The potential savings in road maintenance as well as in vehicle operating costs could be of major significance to both the operator and the community at large, as summarized in the advantages above.

The design of an optimal, software based controller is deemed more complex than the design of the electronic hardware and the mechanical suspension components. However, it is not considered to be a significant obstacle towards the development of active suspension for heavy vehicles.

Recommendations:

1. ACTUAL TESTING OF AN ACTIVE SUSPENSION.

There are two main reasons for this. Firstly, the potential for economic benefits to Canada as a whole through the use of active suspensions is significant (by means of enhanced safety as well as reduced road deterioration). Secondly, a window of opportunity for the development of a Canadian based manufacturer of active suspensions (for the supply of North American assemblers of heavy vehicles) will shortly be emerging.

Based on the results of this study's research, experimental investigation is entirely warranted. It may be presumptuous to firmly assert, without hard proof, that an active suspension will bring about physical benefits, however, the fact that such suspensions can enhance the roadholding of road vehicles as well as reduce mechanical fatigue levels is undeniable. It has already been demonstrated that racing cars equipped with an active suspension can exhibit greater overall tractive capabilities than their passively-sprung counterparts, particularly on roughly paved circuits consisting of city streets (viz. results of 1987 Monaco and Detroit Grand Prix). While the extrapolation of such results to heavy vehicles cannot be made directly, it is reasonable to posit that similar benefits would be obtained by trucks and buses making use of an active suspension.

2. USE OF LESS DAMAGING AND SAFER SUSPENSIONS MUST BE ENCOURAGED AND/OR ENFORCED.

It has been shown by several studies that existing heavy vehicle suspensions possess stability and road reaction force characteristics which are inversely related. An experimental trial of an active suspension will very likely show that an active system can combine low levels of road reactions with high vehicle stability.

In general terms, it is in the direct interest of both the provincial and the federal administrations to encourage the use of suspensions which are less prone to accidents and to the damage of road surfaces. The financial savings on the part of both governments promise to be substantial.

The possible means of encouragement are numerous: policies or programs specifically designed to promote the adoption of such suspensions are an example. On a provincial level, this could include lower road use fees as well as reduced licensing costs for vehicles equipped with an active suspension. Such strategies would be justified by the lower level of road damage costs associated with these types of suspensions.

3. OTHER MEANS OF ENHANCING THE STABILITY AND SAFETY OF HEAVY VEHICLES SHOULD BE INVESTIGATED AND TESTED.

As an interim measure, prior to the general implementation of active suspension systems, other means of enhancing the stability and safety of heavy vehicles should be investigated. Such means could include the review of basic suspension geometries, tire combinations, multi-trailer coupling systems and others, as touched upon in this report.

11. REFERENCES

JOURNAL TITLE	PAGE
Vehicle System Dynamics	65
IMechE Proceedings	68
Automatica	69
IEEE Control Systems Magazine	69
IEEE Vehicle Technology Systems (Convergence IEEE/SAE)	70
Ontario Ministry of Transport and Communications	70
Saskatchewan Highways and Transportation	70
Transport Canada Publications	70
Automotive Engineer (IMechE Journal)	71
RTAC/CTRC Proceedings and Publications	71
ASME Journal of Dynamic Systems, Measurement and Control	72
ASME Publications	72
International Journal of Vehicle Design	73
Highway Research Board Proceedings	73
HSRI Research	73
SAE Papers and Magazine	74
Japan SAE	78
AQTR	78
Commercial Motor	79
ISATA Conference	79
Reinforced Plastics/Composites Institute (S.P.I.)	79

Ingénieurs de l'Automobile 80
L'Information du Véhicule 80
FISITA Congress 80
CAR Magazine 81
Forschungsvereinigung Automobiltechnik (FAT) 81
Australian Road Research 82
Traffic Authority of New South Wales 82
AUTOCAR 82

Vehicle System Dynamics

1986 Vol. 15

1 -Theoretical Limitations in Active Vehicle Suspensions;
Dean KARNOPP (U. of California at Davis), pp. 41-54.

2 -The Application of Linear Optimal Control Theory to the Design of Active Automotive Suspensions;
D.A. WILSON (U. of Leeds, Elec. Eng, England)
and R.S. SHARP (U. of Leeds, Mech. Eng., England)
and S.A. HASSAN (U. of Leeds, Mech. Eng., England)
pp. 105-118.

3 -Lateral Dynamic Behaviour of Truck-Trailer Combinations due to the Influence of the Load;
H.-C. PFLUG (Daimler Benz AG, Stuttgart-Untertürkheim)
pp 155-175.

1985 Vol. 14

4 -Decoupled Active Suspension Design for Improved Ride Quality/Handling Performance;
K.M. MALEK (Mechanical E[illegible]., MIT, Cambridge, Mass.)
and J.K. HEDRICK (idem).
pp. 78-81 (summary).

5 -Design of an Active Suspension for a Passenger Vehicle Model Using Input Processes with Time Delays;
F. FRÜHAUF (Department of Automatic Control, Mechanical Engineering, U. of Paderborn)
and R. KASPER (idem),
and J. LÜCKEL (idem).
pp. 114-120 (summary).

6 -Active Suspensions Classical or Optimal?;
R. A. WILLIAMS (British Rail R. and D. Division).
pp. 127-132 (summary).

7 -Vehicular Energy Losses Associated with the Traversal of an Uneven Road;
X. P. LU (Trans. Research Institute, Ann Arbor, MI.)
and L. SEGEL (idem).
pp. 166-171 (summary).

8 -On the Optimal Ride Control of a Dynamic Model for an Automotive Vehicle System;
P. BARAK (GMI, Flint, Michigan)
and H. K. SACHS (Wayne State U., Detroit, Michigan).
pp. 196-200 (summary).

1984 Vol. 13

9 -Optimal and Suboptimal Linear Active Suspensions for Road Vehicles;
A.G. THOMPSON (Mechanical Eng., MIT, Cambridge, Mass.)
PP. 61-72.

10 -The Chatter of Semi-Active On-Off Suspensions and its Cure;
Donald L. MARGOLIS (U. of California at Davis)
and Mehrnaz GOSHTASBPOUR (idem.).
pp. 129-144.

11 -Adaptive Suspension Concepts for Road Vehicles;
Dean KARNOPP (U. of California at Davis)
and Donald L. MARGOLIS (idem.).
pp. 145-160.

1983 Vol. 12

12 -Dynamic Pavement/Wheel Loading for Trucks with Tandem Suspensions;
M. SAYERS (Transportation Research Institute, Univ. of Michigan, Ann Arbor, U.S.).
and T.D. GILLESPIE (idem.).
pp. 171-172 (summary).

13 -Active Hunting Control of a Wheel/Rail Vehicle Demonstrated up to 530 Km/h on the German Roller Test Rig;
G. DREHER, F. JOBST, L. MAUER, P. MEINKE, A. MIELCAREK
J.NESTMEIER, H. ORLEY (MAN Neue Technologie, Munich).
pp. 173-176 (summary).

14 -Active Controls in Ground Transportation - A Review of the State-of-the-Art and Future Potential;
R.M. GOODALL (Loughborough University of Technology, Leicestershire, U.K.)
W. KORTÜM (Institute for Dynamics of Flight Systems, DFVLR, Oberpfaffenhofen, FRG.).
pp. 225-257.

15 -Active Damping in Road Vehicle Suspension Systems;
Dean KARNOPP (U. of California at Davis).
pp. 291-316.

16 -Semi-Active Control of Wheel Hop in Ground Vehicles;
Donald L. MARGOLIS (U. of California at Davis).
pp. 317-330.

1982 Vol. 11

17 -Semi-Active Heave and Pitch Control for Ground Vehicles;
Donald L. MARGOLIS (U. of California at Davis).
pp. 31-42.

18 -The Response of Active and Semi-active Suspensions to Realistic Feedback Signals;
Donald L. MARGOLIS (U. of California at Davis).
pp. 267-282.

1981 Vol. 10

19 -The Effect of Suspension System Nonlinearities on Heavy Truck Vibration;
T.D. GILLESPIE (University of Michigan, H.S.R.I.) and M. SAYERS (idem).
pp. 97-102 (summary).

20 -The Theory of Stability and Controllability in Consideration of Suspension and Steering System;
HIROSHI HARADA (Toyota Motor Co., Japan), and TOSHIYUKI HASHIMOTO (idem), and ATSUSHI WATARYI (Tokyo University, Japan).
pp. 106-112 (summary).

21 -Self-excited Vibrations of Truck Tandem Axle Suspension and Transmission Systems;
R.S. SHARP (U. of Leeds, England) and C.J. JONES (Automotive Products Co. Ltd, Leamington Spa, U.K.).
pp. 201-202 (summary).

22 -Railway Vehicle Active Suspensions;
J.K. HEDRICK (Mechanical Eng., MIT, Cambridge, Mass.)
pp 267-283.

1980 Vol. 9

23 -Roll Dynamics of Commercial Vehicles;
M.K. VERMA (GM Research Labs, Warren, Michigan) and T.D. GILLESPIE (U. of Michigan, Ann Arbor, MI).
pp. 1-17.

24 -Vehicle Energy Dissipation Due to Road Roughness;
Steven A. VELINSKY (Mech. and Ind. Eng., U. of IL. at Urbana-Champaign, Urbana, IL.) and Robert A. WHITE (idem).
pp. 359-384.

1979 Vol. 8

25 -An Adaptive Control for Vehicle Suspensions;
H. K. SACHS (Wayne State U., Detroit, Michigan).
pp. 201-206 (summary).

1978 Vol. 7

26 -Power Requirements for Traversing Uneven Roadways;
Dean KARNOPP (U. of California at Davis).
pp. 135-152.

1976 Vol. 7

27 -An Active Suspension with Optimal Linear State Feedback;
A.G. THOMPSON (University of Adelaide, Australia).
pp 187-203.

IMechE Proceedings

1986

28 -Aircraft Landing Gears - The Past, Present and Future;
YOUNG, DW (Dowty Rotol Limited), Transport Journal,
Proc. I.Mech.E. Vol 200 No D2, pp. 75-92.

29 -The relative performance capabilities of passive,
active and semi-active car suspension systems;
SHARP, R.S., HASSAN, S.A., Mech. Eng., U. of Leeds,
Proc. I.Mech.E. Vol 200 No D3, pp. 219-228.

1985

30 -Development of ultrasonic height sensor and its
application;
T. FUJISHIRO, H. TANAKA, H. KOBAYASHI, F. SUGASAWA
(Nissan Motor Company Limited, Yokosuka, Japan).
Automotive Electronics, Oct.-Nov. 1985, pp. 349-355.

1984

31 -Passenger comfort - the role of active suspensions;
M.G. POLLARD (R and D Division, British Railways),
and N.J.A. SIMONS (idem).
pp. 161-175.

32 -The application of active suspension to high
performance road vehicles;
P.G. WRIGHT (Lotus International, Norwich, U.K.)
and D.A. WILLIAMS (Cranfield Inst. of Tech., U.K.).
pp. 23-28, Microprocessor in Fluid Power Engineering.

1982

33 -Road Transport - Our Opportunities;
C.D. CERNES (Leyland Bus, Leyland Vehicles Limited).
pp. 149-164.

1981 October

34 -Electronic control unit for passenger car antiskid;
J. GERSTENMEIR (Robert Boxch GmbH, Stuttgart, Germany)
Automotive Electronics, pp. 185-190.

1975

35 -Theoretical Analysis of an Active Suspension Fitted to
a London Transport Bus;
M. COTTERELL (London Transport).
pp. 253-281, John Wiley and Sons.

Automatica

1982
36 -A Class of Active LQC Optimal Actuators;
D. HROVAT (Ford Motor Company.).
pp. 117-119, vol. 18, no. 1.

1986-1980 NO OTHER REFERENCES

IEEE Control Systems Magazine

1985 May
37 -Development of a Custom Microprocessor for Automotive Control;
R.C. BREITZMAN (Electronics Div., Ford Motor Company).
pp. 23-28.

1984-1980 NO REFERENCE

IEEE Vehicle Technology Systems (Convergence IEEE/SAE)

1984

38 -Electronic Optimum Ride Control Handling, Height and Suspension;
SHINZO TOTANI (Nippondenso Co., Ltd., Japan).
pp. III-33 to III-39.

39 -Automatic Implement Tilt Control For Kubota Agricultural Tractors;
Y. KATAYAMA, H. KISAKA, M. KINOSHITA, S. MACHIDA
(Technical Center Farm/Industrial Equ., Kubota, Japan)
pp. X-1 to X-11.

1979

40 -Electronic Control Concepts for Suspension Systems;
R. M. RICHARDSON (Ford Motor Company Ltd., UK)
pp. 194-199.

Ontario Ministry of Transport and Communications

1982

41 -Commercial Vehicle Accident Survey;
M.E. WOLKKOWICZ (MTC)
A.M. BILLING (MTC).

Saskatchewan Highways and Transportation

1982

42 -Heavy Truck Involvement in Traffic Accidents;
A.J. POPOFF (Sask. Highways and Transportation).

Transport Canada Publications

1984

43 -Operating Costs of Trucks in Canada;
Trimac Consulting Services Ltd. (Calgary, Alberta).

AUTOMOTIVE ENGINEER (IMechE Journal)

1987 February/March
44 -TRRL Study of Road Damage by Heavy Vehicles
P. FLINN (reviewed by M. JACOBSON) pp. 51.

1984 February/March
45 -Lotus' Active Suspension;
A. BAKER (Automotive Engineer staff), pp. 56-57.

1979 April/May
46 -The potential for active suspension systems;
H.B. SUTTON (Bolton Institute of Technology, UK)
pp. 21-24.

1979 October/November
47 -Synthesis and development of an experimental active suspension;
H.B. SUTTON (Highbury College of Technology, UK)
pp. 51-54.

1975 December
48 -Vehicle Suspension Design to Reduce Road Damage;
CADY, JR; ASTON, RAH (British Leyland Truck and Bus)
pp. 57-60.

RTAC/CTRC Proceedings and Publications

1986
49 -Vehicle Weights and Dimensions Study: Technical Steering Committee Report;
J.R. PEARSON (RTAC, Ottawa) et al.

1985
50 -Pavement Strain and Deflection Under Truck Axle Loadings - Research Program Description;
J.P. PEARSON (RTAC).

1984
51 -A Survey of the Characteristics of Equipment in the Canadian Trucking Fleet;
R.D. ERVIN (T.R.I., University of Michigan).

1983
52 -Relative Pavement Damage From Various Suspension Types of Commercial Vehicles;
A.T. BERGAN, A.T. PAPAGIANAKIS, B.M. CHURKO, N.A. BURNS.
pp F5-F22, 1983 proceedings.

53 -Universal Standard for Correlation of Road Roughness Measurements;
PONNIAH JOSEPH, T. KHAN, W. PHANG, R.HAAS.

ASME Journal of Dynamic Systems, Measurement, and Control

1986 Vol 108, June

54 -Stochastic Optimal Control of Vehicles with Elastic Body and Active Suspension;
A. HAC (Warsaw Tech. University, Warsaw, Poland).
pp. 106-110.

1985 Vol 107, March

55 -Pneumatic Actuators for Vehicle Active Suspension Applications;
D. CHO (Mechanical Eng., M.I.T., Cambridge, Mass.) and J.K. HEDRICK (idem), pp. 67-72.

56 -An Active Suspension for a Formula One Grand Prix Racing Car; J. DOMINY (Rolls-Royce Ltd., Derby, U.K.) and D.N. BULMAN (R.M.C. of Science, Swindon, U.K.).
pp. 73-78.

1984-1978 NO REFERENCES

ASME Publications

1985 November

57 -The Influence of Suspension Design on Dynamic Pavement Loading;
J. ISAACSON et al (M.I.T, Cambridge, Massachusetts).
Proceedings of "Dynamic Systems: Modelling and Control", ASME November 1985, pp. 333-340.

58 -Two Contrasting Versions of the Optimal Active Vehicle Suspension;
D.C. KARNOPP (U. of California at Davis).
Proceedings of "Dynamic Systems: Modelling and Control", ASME November 1985, pp. 341-346.

1978 December

59 -Are Active Suspensions Really Necessary?;
D.C. KARNOPP (U. of California at Davis).
ASME Paper 78-WA/DE-12.

1975 December

60 -Active Suspensions For Ground Transport Vehicles - A State of the Art Review;
HEDRICK, JK; WOMLEY, DN (MIT, Cambridge, Mass.)
ASME Mechanics and Transportation Systems, pp. 21-39.

1973 September

61 -Transit Car Ride Quality: Prediction, Test and Improvement;
RINEHART, R.E.; ROACH, R.E.; BAIN, J.A.; CROSHAW, P.F.
(Transportation Technology Center, G.E. Transport.)
Intersociety Conference on Transportation, Sept. 23-27
ASME Paper 73-ICT-76.

International Journal of Vehicle Design

1985 Vol 6, No. 4/5

62 -Active and semi-active suspensions for engines and vehicles;
D. L. MARGOLIS (U. of California at Davis)
pp. 571-572 (summary).

63 -Selection of the characteristics of the vertical rigidity of the suspension of a vehicle equipped with an antislip system;
T. KAPITANIAK (U. of Lodz, Ins. Applied Mech., Poland)
pp. 577-579 (summary).

1985 Vol 6, No. 3

64 -Handling performance of truck-trailer vehicles: A state-of-the-art survey;
F. VLK (Dept. Auto. Eng., Tech. U. of Brno, Czech.)
pp. 323-361.

1984 Vol 5, No. 4

65 -Dynamic stability of heavy articulated vehicles;
A.G. NALECZ (New Mexico State U., Las Cruces N.M.),
and J. GENIN (idem), pp. 417-426.

HIGHWAY RESEARCH BOARD PROCEEDINGS

1961

66 -A Method for Introducing Dynamic Vehicle Loads into Design of Highways;
QUINN, BE; VAN WYK, R; KOSSACK, CF (M. Eng. Purdue U.)
HRB Proceedings, 1961 Vol 40, pp. 111-124.

HIGHWAY SAFETY RESEARCH INSTITUTE (HSRI RESEARCH)

1976

67 -Tire Properties and Truck Handling: Some Surprising New Findings;
R.D. ERVIN, C.B. WINKLER.
HRSI Research, July-August 1976, pp. 3-12.

SAE PAPERS AND MAGAZINE

1987

68 -Vehicle-Environment Compatibility with Emphasis on Accidents Involving Trucks; L.E. JACKSON (National Transportaion Safety Board). SAE 870070.

69 -Truck Control Problems Posed by the Design of Highway Ramps; R.D. ERVIN (U. of Michigan), M.A. BARNES (Washtenaw County Road Commission). SAE 870071.

70 -Tire-Pavement Interaction;
R. R. HEGMON (Federal Highway Administration).
SAE 870241

71 -Braking Performance of Heavy U.S. Vehicles;
R.W. RADLINSKI (N.H.T.S.A.) SAE 870492.

72 -Toyota Electronic Modulated Air Suspension for the 1986 SOARER; HARUHIKO TANAHASHI, et al (Toyota Motor Corp.). SAE 870541.

73 -1987 Thunderbird Turbo Coupe Programmed Ride Control (PRC) Suspension; M.W. SOLTIS (Ford Motor Co.).
SAE 870540

74 -Heavy Duty Truck Aerodynamics; R.A. DROLLINGER (Kenworth Truck Co.). SP-688 (February 1987).

75 -Automotive Crash Avoidance Research; (Compendium), SP-699 (February 1987).

1986

76 -Truck Safety - An Agenda for the Future; (Compendium), P-181 (June 1986).

77 -Commercial Vehicle Electronics;
T.O. JONES (TRW, Inc.). SAE lecture SP-647.

78 -Digital Algorithm Design for Wheel Lock Control System
S.F. HUSSAIN, Allied Automotive Bendix Chassis/Brakes
SAE 860509

1985

79 -Heavy Truck Ride;
T.D. GILLESPIE (T.R.I., U. of Michigan). SAE SP-607.

80 -Heavy Truck Safety - What We Know;
H.E. SEIFF (Motor Vehicle Manufacturers Association).
SAE Paper 851191

81 -Rollover of Articulated Vehicles;
P. SWEATMAN, L. MAI (Australian Road Research Board)
SAE Paper 852185

82 -Advanced Semi-Trailer Tractor - Improved Riding Comfort and Simplified Connecting Operations;
T. MORI, B. HAGIWARA, M. SHIRATSUCHI (Hino Motors Ltd)
SAE Paper 852239

83 -An Analysis of Heavy Truck Ride;
Y. YAMADA (Nissan Diesel Motor Co., Ltd.)
and Y. WATANABE (idem.).
SAE Paper 852240

84 -Handling Response and Accident Reconstruction of Tractor-Semitrailers Using Micro and Minicomputers;
J. Ed. MARTINEZ (Texas A&M U., Texas)
and J. G. WALFORD (Verifact Corp., San Antonio).
SAE Paper 852358

1984

85 -1984 Continental Mark VII/Lincoln Continental Electronically-Controlled Air-Suspension (EAS) System;
B.K. CHANCE (Ford Motor Company).
SAE Paper 840342

86 -Active Suspensions with Vibration Absorbers and Optimal Output Feedback Control;
A.G. THOMPSON (Mech. Eng., U.of Adelaide, Australia)
B.R. DAVIS (Electrical/Electronic Eng. U. of Adelaide)
F.J.M. SALZBORN (Applied Mathematics, U. of Adelaide)
SAE Paper 841253

87 -Electronic-Controlled Suspension (ECS);
M. MIZUGUCHI et al (Mitsubishi)
SAE Paper 845051

88 -Development of An Automotive Shock Absorber That Improves Riding Comfort Without Impairing Steering Stability;
N. FUKUSHIMA, M.LIDA, K. HIDAKA.
SAE Paper 845052

89 -Adaptive Suspension Damping;
R.M. RICHARDSON
SAE Paper 845053

90 -Articulated Vehicle Stability - The Role Of Vehicle Design;
P. SWEATMAN, L. MAI
SAE Paper 845055

91 -Comportement Et Tenue de Route Des Ensembles Articulés Routiers: Recherche D'Amélioration Du Seuil De Versement;
P. SORET (Renault, Véhicules Industriels)
SAE Paper 845059

1983

92 -Suspension Design for Optimum Road-Holding;
A.G. THOMPSON (Mech. Eng., U.of Adelaide, Australia)
SAE Paper 830663

93 -The Influence of Size and Weight Variables on the Roll Stability of Heavy Duty Trucks;
R.D. ERVIN (T.R.I., University of Michigan)
SAE Paper 831163

1982

94 -Endurance Testing of Heavy Duty Vehicles;
R.W. MURPHY (Freightliner Corp.).
SAE Paper 820001

95 -The "Marsh-Mellow" Spring;
J.E. CLECK (Central Research Labs, Firestone T.&R. Co)
and J.D. RENSEL (idem)
and D.A. WEITZENHOM (idem).
SAE Paper 820161

1980

96 -An Optimal Linear Active Suspension with Finite Road Preview;
A.G. THOMPSON (Mech. Eng., U.of Adelaide, Australia)
B.R. DAVIS (Electrical/Electronic Eng. U. of Adelaide)
and C.R.M. PEARCE (University of Adelaide).
SAE Paper 800520

97 -Optimum Design of Tractor-Semitrailer Suspension Systems;
M.M. ELMADAY (Wyle Laboratories)
and M.A. DOKAINISH (McMaster University).
SAE Paper 801419

98 -Current Trends in Truck Suspensions; (Compendium),
SP-475 (November, 1980).

1978

99 -Active Ride Control - A Logical Step from Static Vehicle Attitude Control;
M.B. PACKER (Automotive Products Ltd., England).
SAE Paper 780050

100 -Analysis of The Rollover Dynamics of Double-Bottom Tankers;
T.D. GILLESPIE (H.S.R.I., University of Michigan),
and M.K. VERMA (idem),
SAE Paper 781065

1976

101 -Heavy-Duty Truck Suspensions;
E.R. STERNBERG (White Motor Corp.)
SAE Paper 760369

1973
102 -Truck Spring Fundamentals;
SP-376

103 -New Concepts of Tire Wear Measurement and Analysis;
W. BERGMAN (Ford Motor Co.),
and W.B. CRUM (idem).
SAE Paper 730615

1968
104 -Some Fundamental Limitations of Active and Passive
Vehicle-Suspension Systems;
E.K. BENDER (Bolt Beranek and Newman Inc.).
SAE Paper 680750

Magazine

1985 September
105 -Armored combat earthmover can swim, be air dropped;
R.F. HUTTNER (U.S. Army Tank-Automotive Command).
pp. 28-32.

Japan SAE

1986 July
106 -Current Status and Future Trend of Electronic Control Technology Concerning Chassis Components;
N. YOSHIKAWA, T. TANAKA, H. TAKADA, N. KUMAGAI (Mitsubishi Motors Corp., Okazaki-shi), pp. 41-51.

1985 April
107 -The Current Status and Future Trends in Automotive Electronics;
S. MIZUTANI, Y. NAKANO, T. OHTAKE (Nippondenso Co.). pp. 9-19.

108 -Recent Suspension Techniques of Japanese Passenger Vehicles;
Y. KOBAYASHI, M. SHIBUYA, S. ITO, Y. DESAKI (Fuji Heavy Industries, Ltd., Japan), pp. 90-98.

1985 August
109 -Semi-Active Control and Optimum Preview Control Applications to Vehicle Suspension;
K. KAWAGOE (Nissan Motor Co., Ltd., Yokosuka-shi), and M. IGUCHI (Mechanical Eng., University of Tokyo). pp. 24-31.

AQTR

1986

110 -L'entretien routier: priorité de l'heure;
E. KLAESI (ACDI, Section des Infrastructures, Ottawa) Routes et Transports, Automne 1986, Vol. XVI-N°3, pp. 29-31.

111 -Les ponts métalliques...un problème?;
C. HALCHINI, et H. SALAGEANU.
Routes et Transports, Automne 1986, Vol. XVI-N°3, pp. 20-24.

Commercial Motor

1982 March 13
112 -Checking bridge and road wear the German way;
W. BROCK (Commercial Motor staff)
pp. 58-59.

ISATA Conference

1984
113 -Design of Active Suspensions;
G. TOTI (Alfa Romeo Auto S.p.A., Italy),
and F. MARINONI (idem).
ISATA 1984 conference proceedings, paper 84011,
pp. 186-207.

1983-1979 NO REFERENCE

Reinforced Plastics/Composites Institute (S.P.I.)

1986
114 -Composite Leaf Spring for Tank Trailer Suspensions;
K.H. LO, J.J. McCUSKER, W.G. GOTTENBURG, all of
Shell Development Company.
RPC 1986 conference proceedings, Session 12-B,
pp. 1-6.

Ingénieurs de l'Automobile

1986

115 -Sachs: amortissent variable pour montage a posteriori; Octobre-Novembre, pp. 120-121.

116 -Dossier Technique: Etude de la gamme BX Citroën (Les Asservissements Hydrauliques); Juillet-Août, pp. 41-56, Septembre, pp. 45-53.

117 -Deux capteurs de position pour la détection de la hauteur de la voiture;
E. FERRATI, R. GUAGLIUMI, G.P. PERGA (Marelli Autronica), Juillet-Août, pp. 65-68.

118 -A microcontroller with on-chip serial bus interface for automotive applications;
R. ANDRADE (Intel Semiconductor GmbH, Marketing).
Juin, pp. 80-85.

1985

119 -Les systèmes d'amortisseurs, aujourd'hui et demain;
Dr. T. MELLER (Boge GmbH), Octobre, pp. 67-70.

120 -Comportement dynamique de trajectoire d'un véhicule articulé;
M. J.-P. DUPUIS (R.V.I.), Septembre, pp. 65-70.

121 -Suspensions nouvelles réglables pour voitures particulières utilisant les ressorts à air;
M. P. UGAZIO, Octobre, pp. 81-89.

L'Information du Véhicule

1986

122 -L'assistance hydraulique (Direction-Freinage-Suspension) dans le poids lourd et le véhicule de tourisme; Février, pp. 60-72.

FISITA Congress

1978

123 -The dynamic effect of commercial vehicles on the roadway and on the transported goods;
M. STEINER.

CAR Magazine

1986
124 -Re-inventing The Wheel (Lotus Active Suspension); October, pp. 92-99.

1983
125 -Quick Thinker (Lotus Active Suspension); November, pp. 96-101.

126 -Active Racer (Lotus Active Suspension); November, pp. 103-105.

Forschungsvereinigung Automobiltechnik (FAT)

1984
127 -Evaluation of research reports concerning: The influence of vehicle development and design on the road fatigue; October, Published by Union Internationale Des Transports Routiers, Genève.

Australian Road Research

1986

128 -Road User Costs and Charges (Update Insert);
Volume 16(3), Center insert.

129 -Emergency Braking of Semi-Trailer Trucks Using a Belt-Drop Method;
A. HARRISON (CSIRO Division of Applied Physics, Lindfield, Australia)
Volume 16(3), pp.194-198.

130 -Use of Mechano-Lattice Analysis in Prediction of Pavement Performance;
R.R. SMITH (Dept. of Main Roads, Australia)
and W.O. YANDELL (U. of New South Wales, Australia).
Volume 16(1), pp.10-17.

1984 Conference, August
131 -A Study of Heavy Vehicle Stability;
L. MAI (Australian Road Research Board),
and P. SWEATMAN (idem).
Volume 12, part 7, pp.47-56 (10 pages @ $.10).

1982
132 -Vehicular Resistance to Motion as Influenced by Road Roughness and Highway Alignment;
L. SEGEL (Dept. of Mech. Eng., U. of Michigan)
and LU XIAO-PEI (idem).

1974 Conference
133 -Weight Transfer in Trucks;
M.M. CONROY (University of New South Wales),
and G.P. SANDFORD (A.A.R.B)
and N. HARRISON (A.A.R.B).
Volume 7, part 3, pp.60-70.

Traffic Authority of New South Wales

1982
134 -Truck Design, The Shortfalls;
C. W. CORBEN (Traffic Accident Research Unit, TANSW)

1978
135 -Traffic Safety and the Long Distance Truck Driver;
D. R. LINKLATER (Traffic Accident Research Unit, Department of Motor Transport, NSW)

AUTOCAR

1987
136 -My Life At Lotus (Tony Rudd); 22 July 1987, pp. 54-56.

APPENDIX A

HYDRAULIC SUSPENSIONS

Purpose: The purpose of a road vehicle's suspension system is to isolate itself from road surface irregularities and to protect its cargo and/or passengers from harmful acceleration loadings.

Protection Range: A typical passenger vehicle suspension is expected to filter out vertical displacements with amplitudes of 10 cm and frequencies of zero to forty Hertz. Heavy and off-road vehicles are subject to displacements more severe than these.

Traditional Solution: The traditional suspension used to isolate vehicles from bumps and holes has been a spring and damper assembly placed between the road and the vehicle. In most cases this is done by suspending each road wheel on a linkage whose vertical displacements and accelerations are moderated by a spring and a damper.

Spring and Damper Types: Many types of springs can be used, such as steel leafs, coils or bars, a constrained gas, a combination of hydraulic fluid and gas under pressure, rubber in shear, etc. Dampers usually take the form of a hydraulic cylinder whose piston has specially designed orifices which restrain the passage of fluid to a predefined rate according to the piston's velocity and/or acceleration.

Suspension Displacement: The unsprung mass of the wheel is free, within the spring's design limits, to move up or down as it encounters bumps or depressions in the roadway. As this movement occurs, the damper absorbs some of the energy present in the system and prevents the wheel from attaining excessive vertical accelerations.

Spring and Damper Rates: A spring's rate (or characteristic frequency) is usually fixed and consists of a compromise between an effort to maintain an acceptable ride for the occupants (and/or cargo) under certain vertical displacement and load conditions and yet permit safe operation of the vehicle during cornering and other manoeuvers. Dampers can be made to have variable damping rates by using a piston with spring actuated orifices. However, the actual damping range available under such circumstances is limited, and is intended to vary as a function of the vertical velocities and/or accelerations rather than of the displacement itself.

Hydraulic Suspensions: As all modern vehicles use piston type hydraulic shock absorbers (dampers), one might say that they all have "hydraulic" suspensions. However, this terminology is usually reserved for suspensions where both the springing and the damping functions make use of a pressurized fluid (and perhaps a gas) in the actuation and control of wheel reactions.

A hydraulic suspension uses its wheel cylinders as a means of transforming vertical wheel displacements into fluid flow and pressure. This pressure is applied against a compressible or deformable surface, one which exhibits a constant or variable deformation-to-force ratio. In other words, this surface acts as a spring with a linear or non-linear rate. Against atmospheric pressure, such a surface usually depends on hysteresis and/or elasticity for its spring-like action. The Moulton Hydrolastic suspension uses such rubber springs (called "cheeses") which behave in this fashion.

Alternatively, the deformation surface may act against a pressurized gas, as does a flexible diaphragm fixed between a fluid and a gas (oleopneumatic/hydropneumatic system). In this case both the gas and the diaphragm act together as a spring whose rate is variable depending on the initial gas pressure and on the shape the diaphragm takes as it deforms into the gas chamber. Both Citroën and Moulton Hydragas, as well as some Mercedes and Rolls-Royce models, make use of hydropneumatic accumulators as gas springs. In some cases an engine driven pump supplies high pressure hydraulic fluid to the wheel cylinders in order to adjust ride heights.

Damping in a hydraulic suspension is usually accomplished by integral means: a restriction of the fluid's flow on its way from the wheel cylinder to the spring. Usually located at the base of the gas or rubber spring, the damping mechanism consists of a plate with one or more orifices. As opposed to normal practice, such a configuration leads to damping which is in series with the springing rather than in parallel.

Non-Active Hydraulic Suspensions:The Citroën suspension system, an oleopneumatic one, uses mechanical sensors in order to assure a constant ride height and to prevent excessive lateral acceleration roll. These mechanical sensors open or close valves leading from an engine driven hydraulic pump to the wheel cylinders. Damping rates are constant, and spring rates vary in a non-linear fashion due to the way in which the diaphragm deforms into the gas reservoir. A manual control is used to adjust the ride

height according to road conditions: normal height for clear roads, increased height for snow laden or rock strewn roads.

The famous, "pillow" soft ride exhibited by Citroën cars is due to a low effective spring rate. This is possible as the ride height is not affected by the load being carried. In order to compensate for the occasions when high loads are imposed, vehicles equipped with fixed rate springs must have higher initial spring rates. Such high rates are compromised at every load setting other than the prescribed one, and thus contribute to a harsher ride when carrying low loads.

The Citroën system, as sophisticated as it may be, cannot be characterized as an active one. In order to be so, it would have to react to road irregularities by instantaneously varying its damping rate and its spring rate, as well as eliminate most of the roll and pitching effects induced by certain driving maneuvers. This is not the case.

The two Moulton systems, Hydrolastic and Hydragas, are not active either. Their main distinguishing design characteristic is a connection between the front and rear wheel suspensions on the left and right sides of the car. Upon compression of the front wheel's suspension cylinder, part of the displaced fluid acts upon the front spring (rubber or gas). The remaining portion is transferred to the rear cylinder by a hose, where it acts against the load at the rear wheel and also against the rear spring. Essentially what occurs is a redistribution of wheel loadings in an effort to maintain the car parallel to the road and thus reduce motions in pitch.

As the circuit between the two wheels is closed, no additional fluid can be introduced in order to compensate for higher occasional loads: there is no inherent levelling system. A levelling effect may be created by externally increasing the pressure of the gas in both the right and left accumulator type springs, at the front or rear axle, as required. However, this is only applicable to the Hydragas version, and results in a rough approximation of the amount of levelling required. Once the temporary load is removed, the excess pressure must be released, in order to return to a normal ride height.

A secondary effect of the lack of a levelling system in a closed circuit is the change in ride height, spring rate and damping rate which is induced by changes in ambient temperatures. Under Canadian driving conditions, this can result in a winter ride height as much as 3 cm lower than in

summer, the exact opposite of what would be required. In order to minimize seasonal variations in ride height, a greater than desirable spring rate is adopted which compromises to a certain extent the very characteristic for which a hydraulic suspension is usually specified, namely, a low spring rate. Further, neither Moulton system provides antiroll characteristics on its own. Rather, antiroll bars must be installed in order to connect the suspension linkages of wheels on the same axle in order to minimize roll.

Despite the fact that both the Moulton and Citroën systems are not active ones, their ride characteristics are superior to those of conventional suspensions such as fixed rate steel springs and dampers.